Compressor Performance

Aerodynamics For The User

Third Edition

M. Theodore Gresh

Compressor Performance

This page intentionally left blank

Compressor Performance

Aerodynamics for the User

M. Theodore Gresh

ELSEVIER

Butterworth-Heinemann
An imprint of Elsevier

Butterworth-Heinemann is an imprint of Elsevier
The Boulevard, Langford Lane, Kidlington, Oxford OX5 1GB, United Kingdom
50 Hampshire Street, 5th Floor, Cambridge, MA 02139, United States

Library of Congress Cataloging-in-Publication Data
A catalog record for this book is available from the Library of Congress

British Library Cataloguing-in-Publication Data
A catalogue record for this book is available from the British Library

ISBN: 978-0-12-814219-6

Publisher: Matthew Deans
Acquisition Editor: Brian Guerin
Editorial Project Manager: Charlotte Rowley
Production Project Manager: Kamesh Ramajogi
Cover Designer: Matthew Limbert

Typeset by SPi Global, India

Dedication

In memory of my good friend Herman Zijlstra

This page intentionally left blank

Contents

Part II
Application

This page intentionally left blank

Preface

This text has been designed to be used primarily by equipment users, as a guide in selecting, monitoring, and enhancing the aerodynamic performance of various types of compressors. Some basic theory is included as an aid in helping field personnel to better understand the aerodynamics of compressors so that performance enhancements and trouble resolution can be more readily realized. As much as possible, I have attempted to stick to the "business end" of the applicable aerodynamic principles.

This book is the result of various books, articles, notes, seminars, and personal experience that I have collected over the years working in the field of compressor aerodynamics. As it is such a "collection," references have been used extensively as noted.

The concepts and procedures presented in the following pages, while generally in line with Elliott Company Policy and Industry Standards, include opinions belonging solely to me. Conforming to guidelines in this text therefore does not mean compliance with Elliott Company, API, or other industry standards. The methods presented are meant to be guidelines used for day-to-day performance trending or as the first step in selection, troubleshooting, or retrofitting equipment. For potential warranty cases, customer and vendor must agree on a specific test procedure before proceeding. For an "out-of-warranty" problem, the field engineer is best advised to get some help from the equipment manufacturer or 3rd-party consultant, after some initial analysis is completed.

M. Theodore Gresh

This page intentionally left blank

Acknowledgments

A special thanks to Elliott Company for the support in this endeavor and permission to print the various drawings and articles used throughout this book. Thanks also to GHH, Sulzer, Carrier, Dresser Rand, General Electric Co., Ebara, Demag, Ingersol Rand, DuPont, Gulf Publishing, General Engineering Solutions, Mechanical Solutions and Natural Gas Processors Suppliers Association for technical data provided.

Thanks also to all the people who have personally helped by offering their technical support and encouraging comments, with special thanks to Marc Sassos, Dave Galster, Don Rudisel, Ron Aungier, John Beaty, Bob Spigarelli, Paul Gallick, Bill Hohlweg, Katsuhiko Yamanaka, Dennis Maffessanti, Ross Hackel, Kathy Lazur, Bob DeMaria, James Sorokes, Steve Ross, and my wife, Lynn.

The talents of Frank Weidler, Gerry Brunson, Tom Humphrey, Pawel Kapelanczyk, John Holland, Mike Wieliczki, and Ed Bennett are displayed throughout this book in the various graphics they created for me.

This page intentionally left blank

Symbols

Symbols

A	area, ft^2
a	speed of sound, ft/s
BHP	brake or shaft horsepower
C	discharge coefficient
c_p	specific heat at constant pressure, BTU/lb mole °R
c_v	specific heat at constant volume, BTU/lb mole °R
D	pipe diameter, in.
d	throat, or impeller diameter, in.
E	voltage
E	velocity of approach factor
Eff	efficiency
Fa	orifice meter thermal expansion factor
g_c	Gravitational constant $32.2 \frac{\text{ft-lb mass}}{\text{lb force-s}^2}$
GHP	gas horsepower
H	Head $\frac{\text{ft-lbs force}}{\text{lb mass}}$
HP	horsepower
h	enthalpy (BTU/lb mass)
h_w	differential pressure, inches water
I	amperage
K	flow meter flow coefficient
k	adiabatic exponent (c_p/c_v)
MW	molecular weight
$\dot{M}$	weight flow (lb/min)
M	Mach number, V/a
N	speed, RPM
N_s	specific speed
n	polytropic exponent
P	static pressure (psia)
P_c	critical pressure (psia)
P_r	reduced pressure
P_T	total pressure, psia
P_0	stagnation pressure, psia
P_v	velocity pressure
PF	power factor
Q	flow rate, ft^3/min
Q_s	flow rate, ft^3/s
q	heat transfer, ft-lb force/lb mass
R	gas constant (1544/MW)
Re	Reynolds number
r_p	pressure ratio (P_2/P_1)
s	entropy, BTU/°F/lb
SHP	shaft horsepower
T	absolute temperature (°Rankine = °F + 459.6)
T_c	critical temperature (°Rankine)
T_R	reduced temperature (T/T_c)
t	temperature (°F)
U	tip speed, FPS
u	internal energy, ft-lb force/lb mass
V	velocity (ft/s)
v	specific volume (ft^3/lb mass)
W	Work $\frac{\text{ft-lbs force}}{\text{lb mass}}$
Y	flow meter expansion factor
Ya	adiabatic expansion factor
Z	compressibility factor
z	vertical height

Greek Letters

β	throat (or orifice) to pipe diameter ratio
η	efficiency
γ	work coefficient
μ	head coefficient
μ'	absolute viscosity, lb-s/ft^2
v'	kinematic viscosity, ft^2/s
ρ	density, lb/ft^3
ϕ	flow coefficient

Subscripts

ad	adiabatic process (H_{ad})
p	polytropic process (H_p)
S	standard conditions—usually 14.7 psia, 60°F, dry air
1	inlet conditions (P_1)(Q_1)(t_1)
2	discharge conditions (T_2)(P_2)

This page intentionally left blank

Part I

Theory

This page intentionally left blank

Chapter 1

Introduction to Aerodynamics

Down through the years, human needs and desires have required a continued evolution of more and more sophisticated fluid-handling apparatus. In general, fluid handling involves two problems, fluid transportation and fluid pressurization.

Ancient man was most concerned with liquid transport and storage. Of primary concern was irrigation for agricultural purposes and transport of water to cities.

The Bronze Age, which began about 3000 B.C., brought with it the requirement of mechanisms for enhancing air supply to hearth furnaces. Air was first introduced in hearths by crude drafts and simple fanning. With time, innovation brought improved air supply devices. Hearths were oriented to capture the prevailing winds, and chimneys were added to help draw more air to the furnaces.

With the advent of the Iron Age, which began around 1000 B.C., no longer were simple drafting techniques adequate. A much higher hearth temperature required a pressurized air blast. Small foot- and hand-operated bellows were used in the small hearths of the farrier and blacksmith. Five hundred years ago, immense bellows were used in Germany to supply the air required for large furnaces. These were ultimately supplemented by piston pumps. Today, rotary compressors are used for this purpose.

The Industrial Revolution and, most recently, the Space Age, have produced an exponential growth in the advancement of turbomachinery, from the simple squirrel cage fan in a car's heater to the liquid fuel pumps used on the space shuttle engines.

FLUID MECHANICS AND THERMODYNAMICS

Little heed was paid to the various fluid properties in the design of compression devices until the 19th century. Until this period, only a slight density and temperature change was encountered at the reduced compression ratios used in air pumps. The designer had a large margin of error possible since he was at liberty to "tinker" and adjust the apparatus at the job site until it was perfected. In most instances, both the building and design were done at the job site.

Concepts of flow, energy, work, heat, and momentum, which eluded the grasp of the early Greek philosophers and later the Roman engineers, gradually began to be understood and interpreted under the impetus of the Renaissance scientists da Vinci, Galileo, Newton, Bernoulli, Euler, St. Venant, Stokes, and Navier. The mathematical tools to describe and solve problems were wrought by Liebniz, Newton, De Moivre, Descartes, Legendre, and others. Watt, Stephenson, Carnot, Clausins, and Thurston through their applied efforts on the steam locomotive developed technical, mechanical, and thermodynamic solutions, which have contributed to the compression equipment of our century. The science of heat transfer, thermodynamics, and energy conservation was developed by Maxwell, Thurston, Otto, Helmholtz, Steffan, Boltzmann, Rayleigh, Rankine, Mach, and Plank. In the wake of the Wright Brothers' first flight at Kitty Hawk came the aerodynamic scientists Kutta, Joukowsky, Von Karman, Von Mises, Prandtl, Lamb, Struhal, Tiejens, Stodola, Dryden, Parsons, and Paulson. With the advent of flight, these men developed theories on boundary layer, vortex shedding, aeroelastic phenomena, and other necessary tools used in the design of present-day turbomachinery [1].

FIRSTS

In Alexandria, Egypt, about 130 A.D., a priest scientist named Hero employed aerothermo principles to generate steam and drive a small reaction turbine.

Although the fluid mechanics of a compressor and turbine are much the same, knowledge of fluid mechanics is much more crucial for the design of a compressor than for a turbine. A turbine, with its flow usually going from a high to a low pressure, will always work. With reasonable design, it will work at a respectable efficiency. A compressor, conversely,

Compressor Performance. https://doi.org/10.1016/B978-0-12-814219-6.00001-5

particularly an axial compressor, will not produce any pressure rise at all unless properly designed. Consequently, very little activity was seen in the field of compressor design until the 18th century.

In 1705, Denis Papin published full descriptions of the centrifugal blowers and pumps he had developed; however, the efficiency of these machines is unknown [2,3].

John Barber designed and patented a gas turbine engine in England in 1791. The engine was designed to operate on a constant pressure cycle using gas from wood or coal as fuel [4].

In 1851, Henry Gifford flew from Paris to Trappes in the first successful aircraft propulsion device, a propeller-driven dirigible balloon powered by a steam engine [5].

In 1872, Dr. Stolze patented a gas turbine, which was eventually built and operated. The engine employed a multistage axial-flow compressor and a multistage turbine with both mounted on the same shaft. Heat was supplied to the air by means of a furnace located between the compressor and turbine [4].

Around the same period, Parsons and Delaval developed a reaction steam turbine, for the purpose of driving blowers and generators. Although Parsons also used this device in reverse to serve as a compressor, the efficiency was low—around 60%. Sir Charles Parsons' 1884 patent also made reference to the gas turbine engine and provided for cooling to the turbine blades [1–3].

The first United States patent covering a gas turbine was by Charles Curtis (inventor of the Curtis steam turbine) in June of 1895 [3].

In 1905, Dr. Alfred J. Buchi of Switzerland first suggested the turbocharger for enhancing the output of internal combustion engines. He later went on to patent his ideas in 1915 and to organize the Buchi Syndicate in 1927 for the purpose of developing his systems [3].

It was not until January 16, 1930, that Frank Wittle, an officer in Great Britain's Royal Airforce, developed and patented a practical design for an aircraft gas turbine engine. However, the British Air Ministry dismissed the design, finding it impractical [3,6].

A few years later in 1934, a German named Hans von Ohain began development of an engine of similar design. In 1936, he joined forces with Ernst Heinkel, an airplane manufacturer. Progress was good and an aircraft with von Ohain's engine was successfully flown in August 1939. Von Ohain's HES8A Engine had a centrifugal compressor and a mixed-flow expander [2,6].

Meanwhile, Wittle had obtained some money from the British Air Ministry to develop his engine. In May 1941, an aircraft with Wittle's jet engine was successfully flown. Wittle's W2/700 Turbojet Engine, which consisted of an axial compressor, a single-stage centrifugal compressor, and an axial expander, was eventually developed into the Rolls-Royce Welland in England and also the General Electric J33 in the United States [3,6].

DEFINITION OF COMPRESSOR

A compressor is a device that transfers energy to a gaseous fluid for the purpose of raising the pressure of the fluid as in the case where the compressor is the prime mover of the fluid through the process. The purpose may also include a desired temperature rise to enhance the chemical reaction in the process.

Devices that develop less than 5.0 psig, or that effect a 7% density increase from inlet to discharge, are classified as fans or blowers. Above this level, the devices are referred to as compressors. Due to the low density change, fan equations assume constant density, thus simplifying the calculations [7,8].

Pumps are very similar to compressors but deal primarily with incompressible hydraulic fluids, whereas compressors generally deal with compressible gaseous fluids.

TYPES OF COMPRESSORS

The two basic types of compressors are positive displacement and dynamic.

Positive Displacement Compressor

The positive displacement compressor functions by means of entrapping a volume of gas and reducing that volume, as in the common bicycle pump, and the screw compressor shown in Fig. 1.1. The general characteristics of the positive displacement compressor are constant flow and variable pressure ratio (for a given speed).

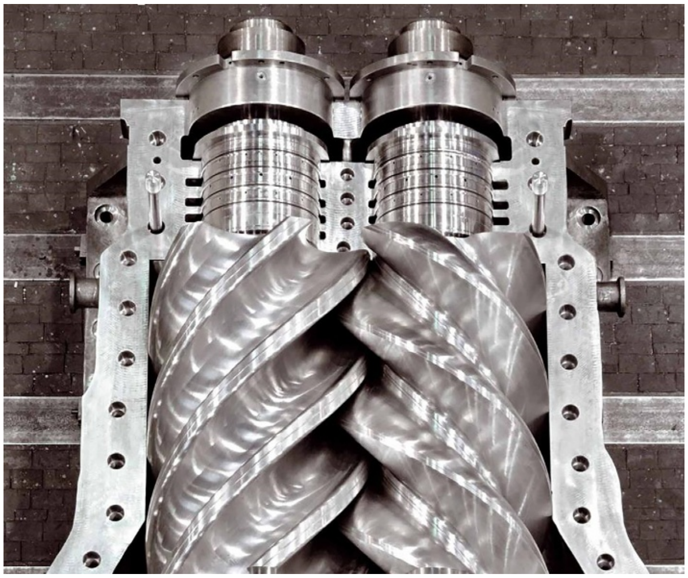


FIG. 1.1 Positive displacement screw compressor. *(Courtesy of MAN GHH.)*

Positive displacement compressors include

- piston compressor
- screw compressor
- vane compressor
- lobe compressor

Dynamic Compressor

The dynamic compressor depends on motion to transfer energy from the compressor rotor to the process gas. The characteristics of compression vary depending on the type of dynamic compressor and on the type of gas being compressed. The flow is continuous. There are no valves and there is no "containment" of the gas, as in a positive displacement compressor. Compression depends on the dynamic interaction between the mechanism and the gas.

Dynamic compressors include

- ejector
- centrifugal compressor
- axial compressor

Ejector

An ejector is a very simple device, which uses a high-pressure jet stream to compress gas. The momentum of the high-pressure jet stream is transferred to the low-pressure process gas. This type of compressor is commonly used for vacuum applications.

Centrifugal Compressor

A centrifugal compressor acts on a gas by means of blades on a rotating impeller. The rotary motion of the gas results in an outward velocity due to centrifugal forces. The tangential component of this outward velocity is then transformed to pressure by means of a diffuser.

Fig. 1.2 is typical of a single-stage centrifugal compressor. A high-pressure multistage compressor is shown in Fig. 1.3.

Axial Compressor

An axial compressor imparts momentum to a gas by means of a cascade of airfoils. The lift and drag coefficients of the airfoil shape determine the compressor characteristics. Fig. 1.4 shows a typical axial compressor. An axial compressor incorporated in a turbocharger is shown in Fig. 1.5.

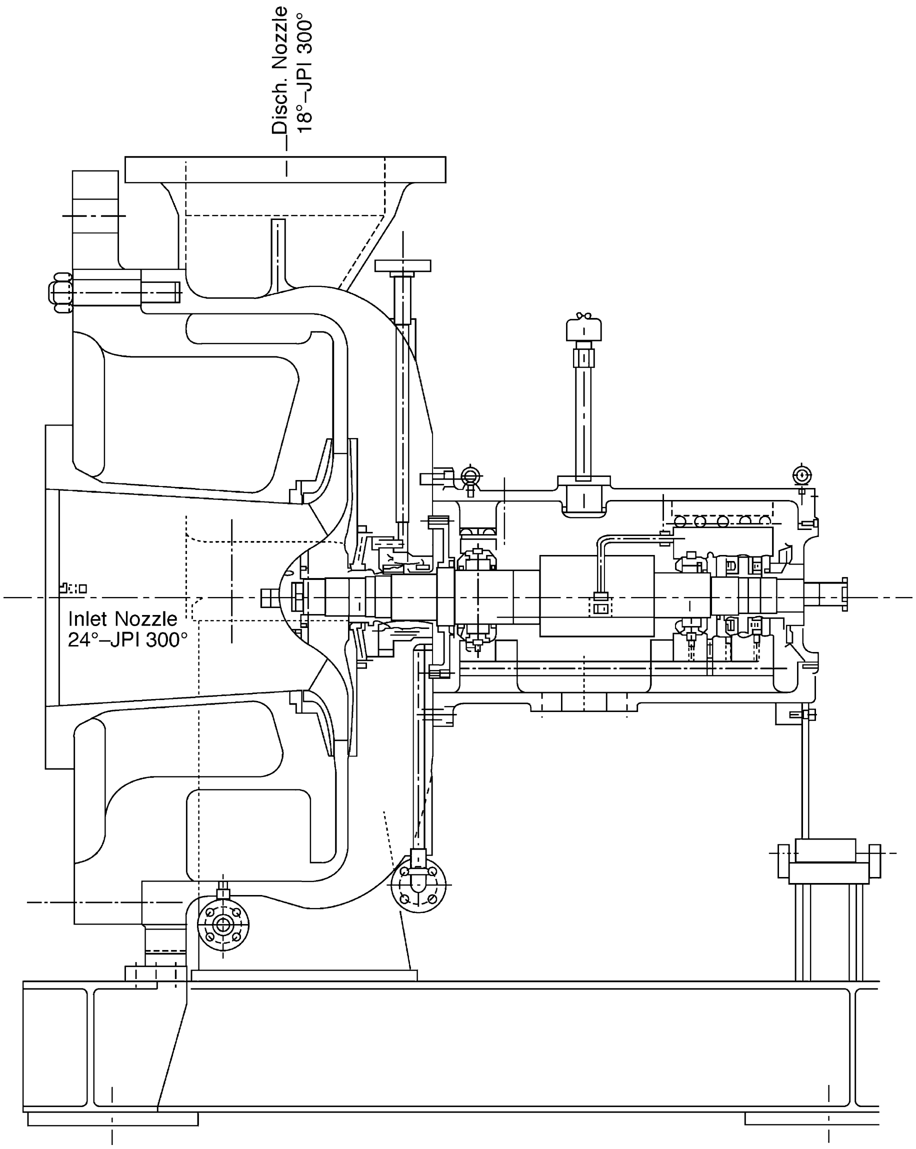


FIG. 1.2 Single-stage centrifugal compressor. *(Courtesy of Ebara Corporation.)*

RELATIVE COMPARISONS OF VARIOUS COMPRESSOR TYPES

Capacity

Axial compressors have the greatest capacity for a given volumetric size. The design is a very compact, light-weight compressor, which can handle a large volume of gas. This explains its popularity for use in aircraft.

Efficiency

Fig. 1.6 illustrates the relative nominal efficiencies for the various types of compressors.

For small capacities, the positive displacement compressor is generally the best. At higher capacities, valve and seal leakage, mechanical friction, and flow discontinuities increase rapidly, limiting overall efficiency.

In a centrifugal compressor, the opposite is true. In small capacities, the sealing surface is large in comparison to the compression element, the impeller. As the compressor size increases, the seal leakage rate grows slowly relative to volume through-put. Reduced mechanisms (bearings, valves, seals) and improved through-flow contribute to improved efficiencies at the high capacities.

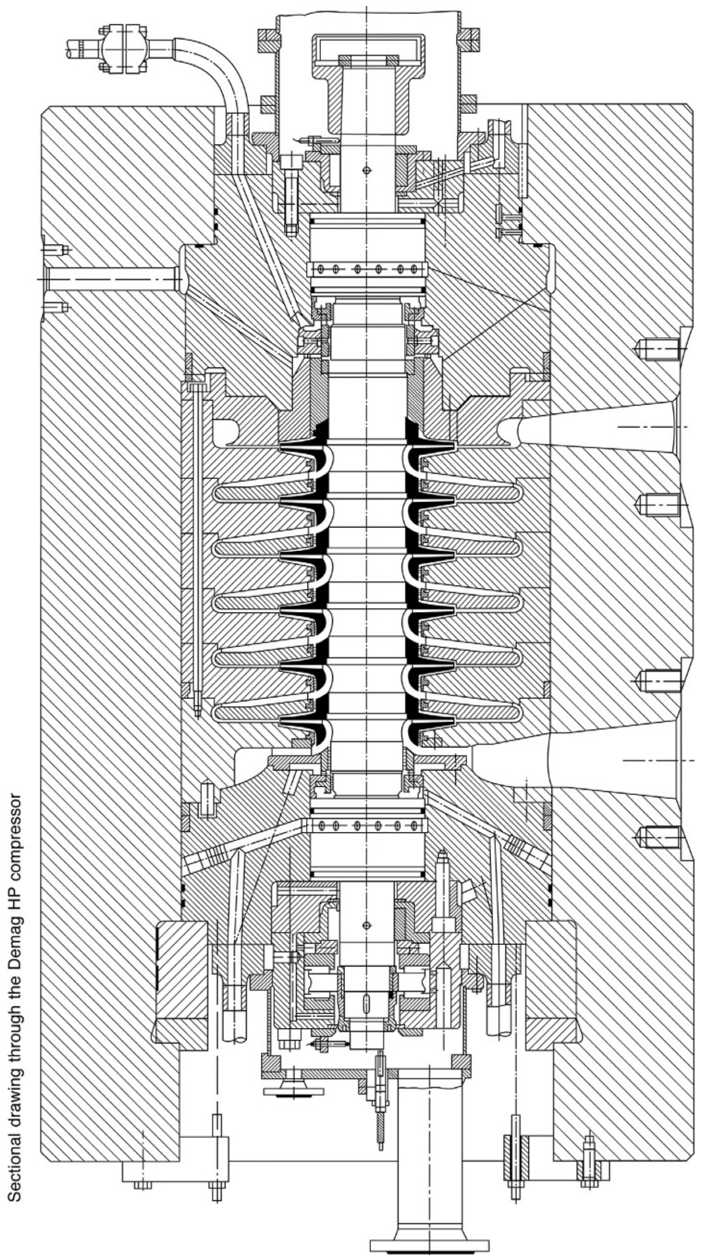


FIG. 1.3 High-pressure barrel-type multistage centrifugal compressor. *(Courtesy of Mannesmann Demag.)*

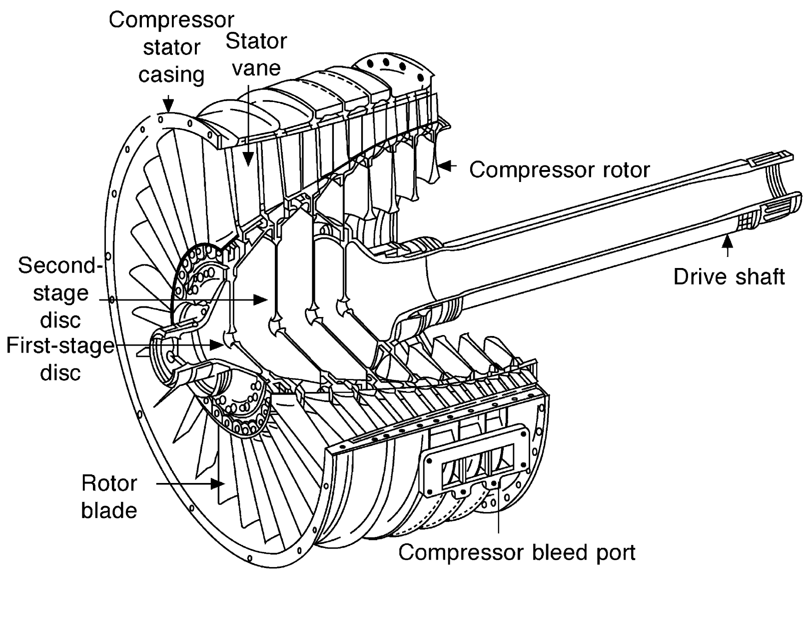


FIG. 1.4 Axial compressor. *(Courtesy of General Electric Company.)*

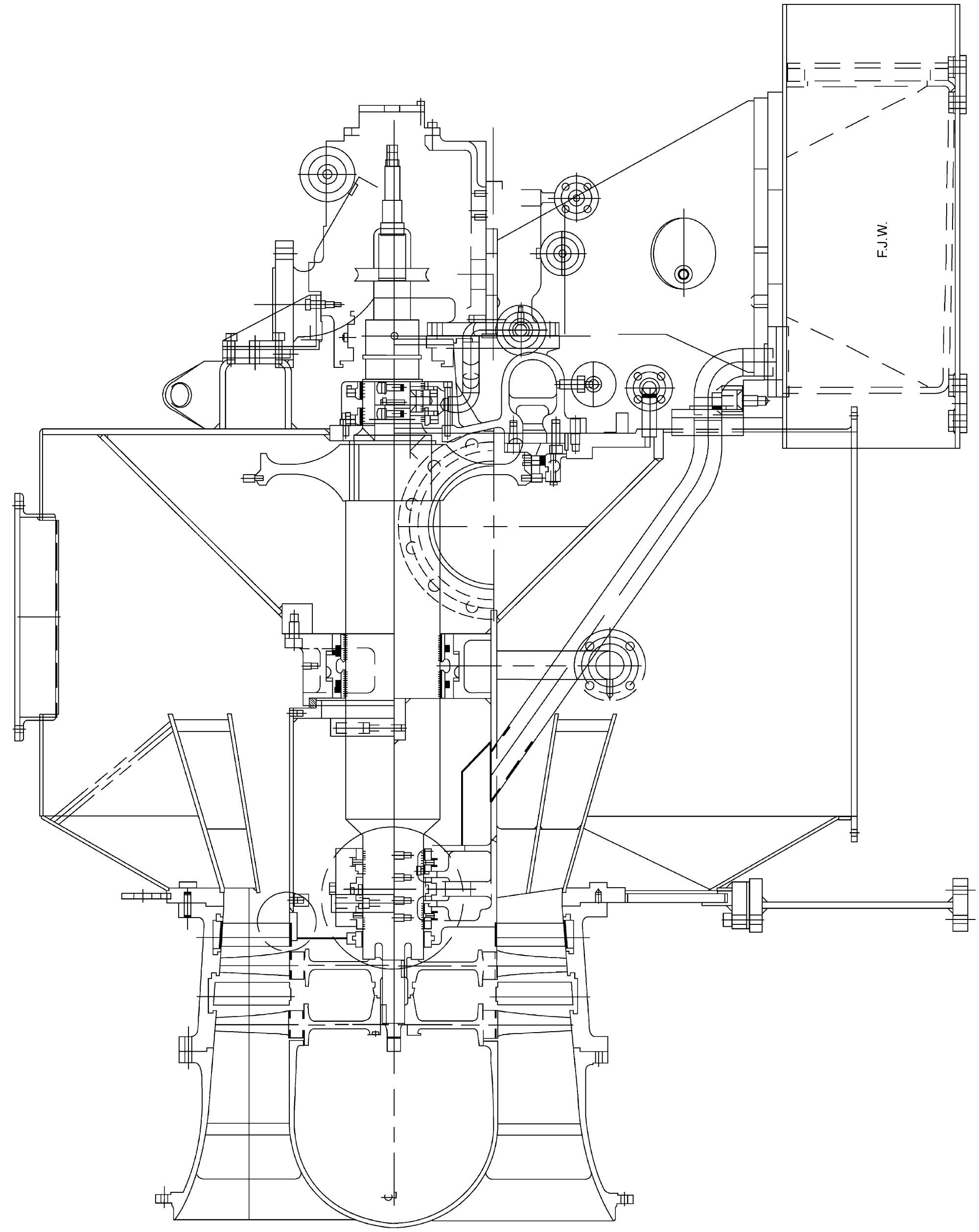
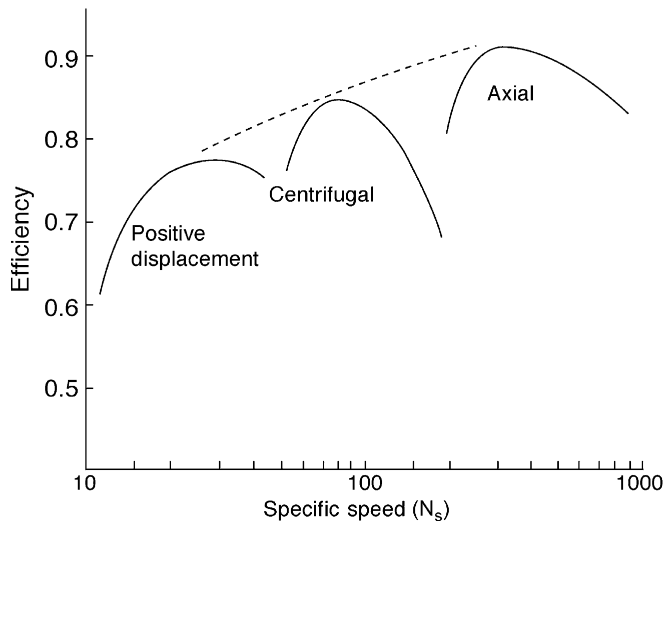


FIG. 1.5 Turbocharger with axial compressor. This unit was used to supply air to a steam boiler unit. *(Reprinted with permission of Elliott Company, Jeannette, PA.)*

FIG. 1.6 Efficiency versus type of compressor. *(Adapted from Sheperd DG (Cornell University). Principles of turbomachinery. New York: Macmillan; 1956.)*



Axial compressors have the best efficiency. Both mechanical and aerodynamic losses for an axial compressor are very low, resulting in efficiencies approaching 90% or even better.

Due to the configuration of the axial, the sealing surface is very small in comparison to the volume of gas flow. Also, the "wetted perimeter" (frictional surface versus the volume flow) is very small, contributing to low losses and high efficiency for large capacities. Further improvement is via the constant nominal through-velocity. Losses due to acceleration and deceleration are limited.

The boundaries shown in Fig. 1.6 are constantly being changed by enhancements to compressor design such as abradable seals to reduce leakage, low friction bearings and seals, and hybrid compressor elements.

Pressure Ratio

Positive displacement compressors and ejectors can have a very high pressure ratio. For dynamic compressors, the centrifugal compressor achieves the highest per stage pressure ratio. Axial compressors develop very low pressure ratio per stage, thus the need for many stages.

Operation

Specialized training is required to operate both centrifugal and axial compressors. The primary concern is to avoid operation in aerodynamically unstable regions, including surge, rotating stall and choke. Operation in these areas can cause equipment damage.

Characteristic Curves

Fig. 1.7 shows the normal characteristic curve shapes for the various compressor types. Positive displacement compressors exhibit a constant volume and a variable pressure ratio, while low-speed centrifugal compressors approach being constant pressure ratio and variable volume machines. Axial compressor characteristics are somewhere in between. Very high-speed centrifugal compressors may have characteristics approaching those of axials.

An in-depth understanding of aerodynamics is required to properly design, select, operate, and maintain centrifugal and axial compressors. The advantages and disadvantages of each type are listed in Table 1.1. This table, along with the following discussions, will provide an overview of aerodynamics for these compressors. The goal is to provide the user with a better understanding of how to more efficiently operate and maintain dynamic compressors. These same tools will be useful in troubleshooting, rerating, and selecting new equipment. Figs. 1.8–1.10 show some other centrifugal compressor styles.

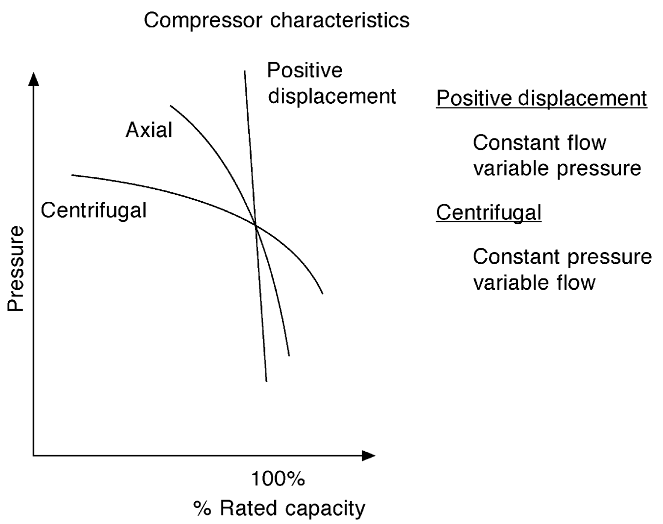


FIG. 1.7 Characteristic curves. *(Data from Paluselli DA. Basic aerodynamics of centrifugal compressors. Jeannette, PA: Elliott Co.)*

TABLE 1.1 Relative Comparison of Compressors

Type	Advantages	Disadvantages
Centrifugal	Wide operating range Low maintenance High reliability	Unstable at low flow Moderate efficiency
Axial	High efficiency High-speed capability Higher flow for given size	Low pressure ratio per stage Narrow flow range Fragile and expensive blading
Positive displacement	Pressure ratio capability not affected by gas properties Good efficiencies at low specific speed	Limited capacity High weight-to-capacity ratio
Ejector	Simple design Inexpensive No moving parts High-pressure ratio	Low efficiency Requires high-pressure source

(Adapted from Paluselli DA. Basic aerodynamics of centrifugal compressors. Jeannette, PA: Elliott Co.)

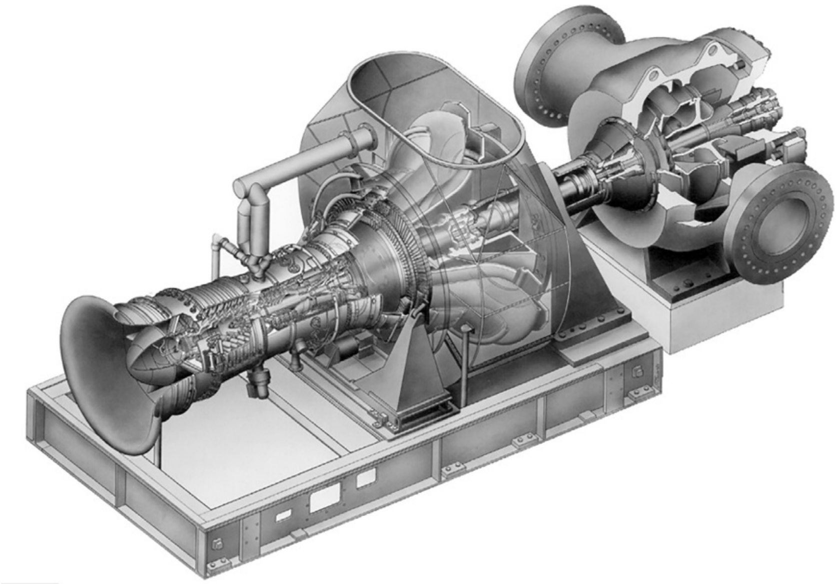


FIG. 1.8 Rolls-Royce RFBB-36 centrifugal pipeline compressor driven by a Rolls-Royce RB211 DLE axial gas turbine. *(Printed with permission of Rolls-Royce Energy Business, Mount Vernon, Ohio.)*

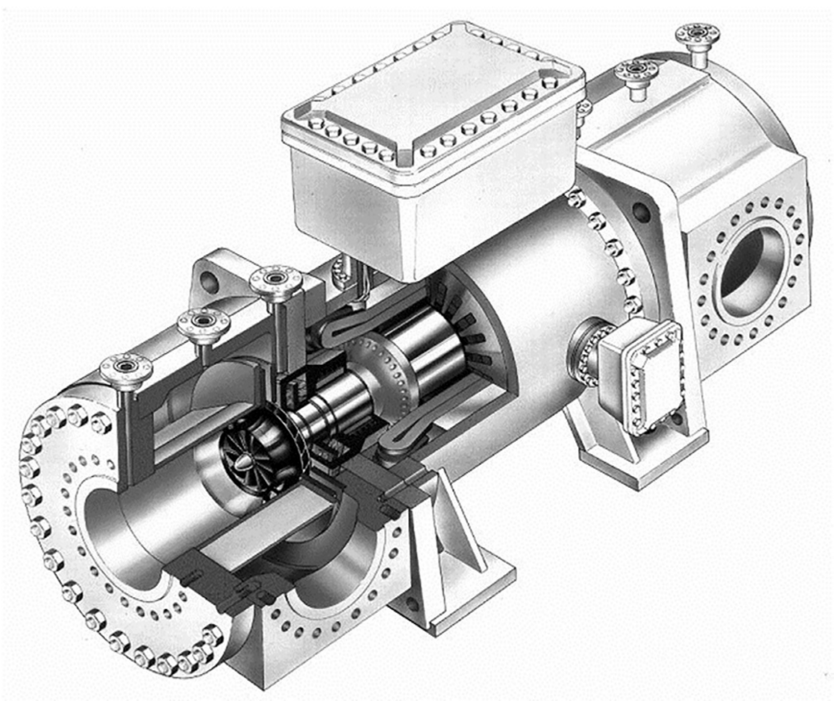


FIG. 1.9 This "MOPICO" centrifugal compressor, designed for pipeline compressor stations, is characterized by its axial and radial magnetic bearings, gas face seals, and high frequency motor, which powers the compressor without a gear box. *(Courtesy of MAN Turbomaschinen AG Schweiz.)*

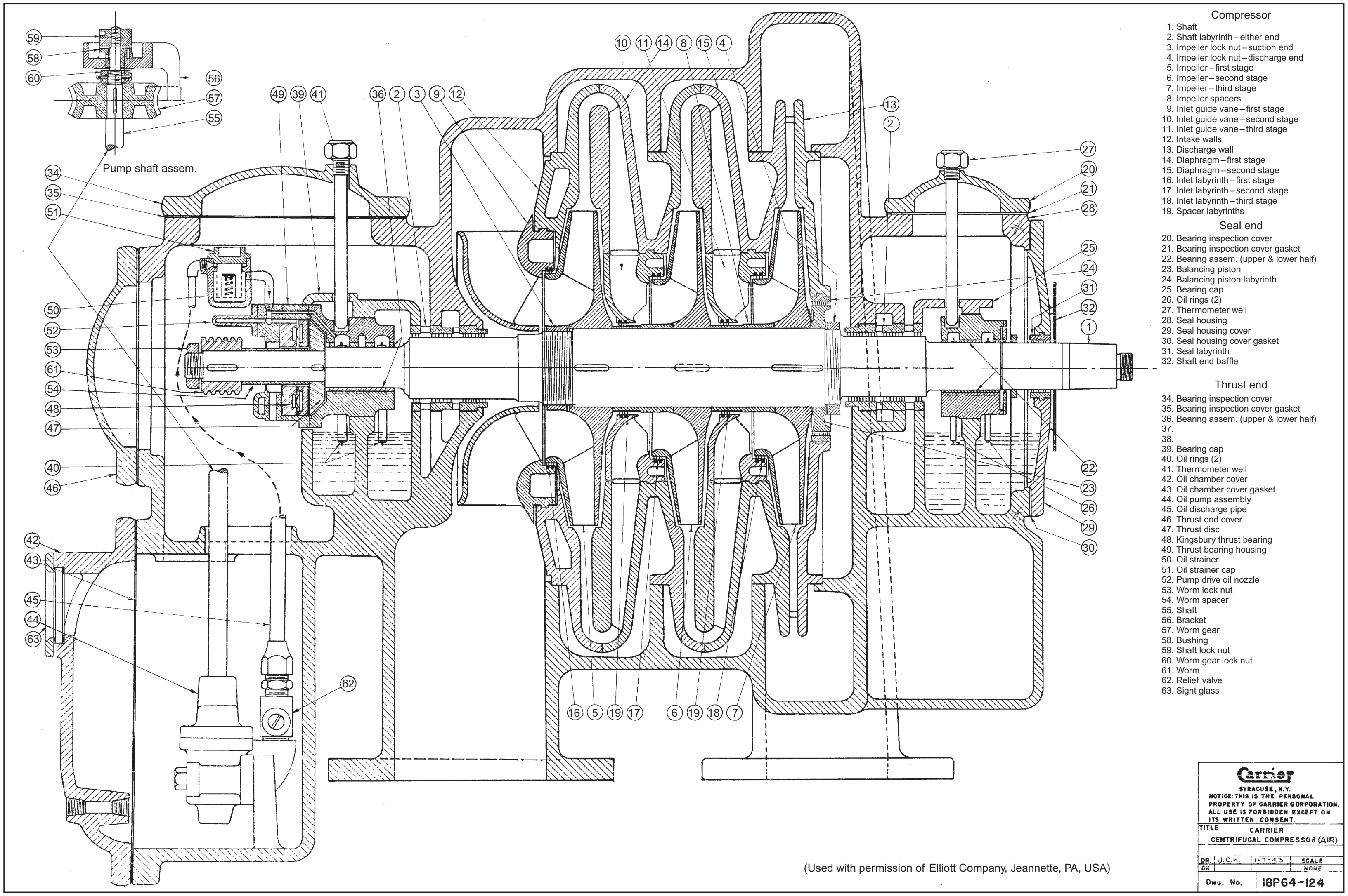


FIG. 1.10 This centrifugal compressor was built by Carrier Corp in 1943 for refrigeration service. Note the shaft-driven oil pump and ring lubricated journal bearings.

This page intentionally left blank

Chapter 2

Thermodynamics

As noted in Chapter 1, knowledge of fluid mechanics and thermodynamics is very important in the proper design of a centrifugal compressor, especially if good efficiency is desired. This knowledge is especially important for the design of an axial compressor. This chapter provides a summary of many of the basic laws and equations used in the design, selection, and operation of centrifugal and axial compressors.

GAS LAWS[1]

No gas conforms exactly to the ideal gas laws. However, most gases conform to these laws with sufficient accuracy to yield good engineering answers. These laws, therefore, are used to form the foundation of compressor thermodynamics.

Boyle's Law

The P_v product remains constant at constant temperature.

$$p_1 v_1 = p_2 v_2, \quad T = \text{constant} \tag{2.1}$$

Charles' Law

The volume of a gas varies directly as the absolute temperature at constant pressure.

$$\frac{v_2}{v_1} = \frac{T_2}{T_1}, \quad P = \text{constant} \tag{2.2}$$

Dalton's Law

In a mixture of gases, the summation of partial pressures equals the total pressure of the mixture.

Avogadro's Law

All gases have the same number of moles in the same volume at the same pressure and temperature.

$$\frac{Pv}{T} = \text{constant} \tag{2.3}$$

The Ideal Gas Law

From Charles' and Boyle's laws

$$v = \frac{RT}{P} \tag{2.4}$$

or

1. Adapted from "Compressor refresher," Elliott Company, Jeannette, PA, 1975, with permission [11].

Compressor Performance. https://doi.org/10.1016/B978-0-12-814219-6.00002-7

$$\rho = \frac{P}{RT} \tag{2.5}$$

where

v = specific volume, ft^3/lb
ρ = density, lb/ft^3
$R = R_0$/MW = 1545.32/MW, ft-lb$_f$/lb-mole °R
R_0 = universal gas constant
= 1.98587 BTU/lb-mole °R
= 1545.32 ft-lb$_f$/lb-mole °R
= 8.3143 joules/gm-mole °K
= 10.73 psia-ft^3/lb-mole °R
MW = gas molecular weight

Most gases at low pressures act as ideal gases.

Note that for a specific gas, the ideal gas equation reduces to

$$\frac{P_1 v_1}{T_1} = \frac{P_2 v_2}{T_2} \tag{2.6}$$

Compressibility

Most gases encountered in industrial compression do not follow the perfect gas equation of state exactly but differ in varying degrees [11]. The degree in which any gas varies from the ideal is expressed by a factor (compressibility), which modifies $Pv = RT$ to

$$Pv = ZRT \tag{2.7}$$

or

$$Z = \frac{Pv}{RT} \tag{2.8}$$

There have been many equations and charts used to determine the value of Z for any given state point. The chart most commonly used has been the Obert-Nelson chart, which in modified form appears in the section on Gas Properties in Appendix A, Fig. A.1.

In order to use the compressibility chart, reduced pressure, P_R, and reduced temperature, T_R, must be determined. These values are found by dividing the state point under consideration by the respective critical values of pressure or temperature. Values for commonly used gases are listed in Appendix A, Table A.1.

The best way to deal with any gas mixture, but especially gases that significantly vary from the ideal gas laws, such as heavy hydrocarbons, more complex real gas equations of state such as Benedict, Webb & Rubin (BWR), as used in Gas Flex, Lee Kesler, as used in CompAero, or others must be used [35].

BERNOULLI'S EQUATION

For incompressible flow

$$Pv + Z + \frac{V^2}{2g} = \text{constant} \tag{2.9}$$

This equation says that the sum of the pressure energy, potential energy, and kinetic energy is constant [12]. It is useful for a system where there is no work done, no heat exchanged, and therefore no change in internal energy such as for frictionless flow through a nozzle. The terms in this equation are in units of length and are frequently called pressure, elevation, and velocity head. Between two separate points in a flow

$$P_1 v + Z_1 + \frac{V_1^2}{2g} = P_2 v + Z_2 + \frac{V_2^2}{2g} = \text{constant} \tag{2.10}$$

Because gases are compressible, this equation is never exactly correct, but in many cases the change in density is insignificant or can be compensated for.

For determining the total or stagnation pressure P_0, $V_2 = 0$,

$$P_1 v + \frac{V_1^2}{2g} = P_2 v \tag{2.11}$$

or

$$P + \frac{\rho V^2}{2g} = P_0 \tag{2.12}$$

The term $\rho V^2/2g$ is called the dynamic or velocity pressure (or head). The sum of the static and dynamic pressures is equal to the total or stagnation pressure for an ideal gas in incompressible flow.

Modified Bernoulli Equation

Adding friction to Bernoulli's equation, represented as head loss between points 1 and 2,

$$\left(P_1 v + Z_1 + \frac{V_1^2}{2g}\right) - \left(P_2 v + Z_2 + \frac{V_2^2}{2g} = H_{1-2}\right) \tag{2.13}$$

For horizontal flow at constant velocity

$$Z_1 = Z_2$$
$$V_1 = V_2$$

therefore

$$H = (P_1 - P_2)v \tag{2.14}$$

The General Energy Equation

By conducting a complete energy balance around a system, we obtain the General Energy equation:

$$\begin{aligned} &q + u_1 + P_1 v_1 + \frac{V_1^2}{2g} + Z_1 = W + u_2 + P_2 v_2 + \frac{V_2^2}{2g} + Z_2 \\ &h = u + Pv \\ &q + h_1 + \frac{V_1^2}{2g} + Z_1 = W + h_2 + \frac{V_2^2}{2g} + Z_2 \end{aligned} \tag{2.15}$$

Assume: q (heat transfer) $= 0$, and negligible elevation and velocity effects.

$$W(\text{work}) = h_1 - h_2 \tag{2.16}$$

THERMODYNAMIC RELATIONS FOR A PERFECT GAS

For a perfect gas [12]

c_v = specific heat at constant volume

$$= \frac{du}{dT}$$

c_p = specific heat at constant pressure

$$= \frac{dh}{dt}$$

By definition

$$h = u + Pv \tag{2.17}$$

or, since $Pv = RT$

$$h = u + RT$$

Therefore:

$$c_p = \frac{d(u+RT)}{dT} \\ = cv + R \tag{2.18}$$

or

$$R = c_p - c_v \tag{2.19}$$

also

$$k = \frac{c_p}{cv} \tag{2.20}$$

Combining these equations,

$$c_p = \frac{k}{k-1}R \tag{2.21}$$

and

$$c_v = \frac{R}{k-1} \tag{2.22}$$

Adiabatic Process

An adiabatic process is defined as a process in which no heat transfer takes place. This does not mean that the temperature is constant, but rather that no heat is transferred into or out from the system. In compressor theory, the terms adiabatic (no heat transfer) and isentropic (constant entropy) are used interchangeably. This is quite valid for the context in which they are used. (The actual definition of an isentropic process is an adiabatic, reversible process.)

For an adiabatic process [12],

$$c_v dT = -P dv$$

and

$$c_p dT = v dp$$

Combined,

$$\frac{dP}{P} = -\frac{c_p}{c_v}\frac{dv}{v} = -k\frac{dv}{v}$$

For k = constant,

$\ln P = \ln v^{-k} + \ln \text{constant}$

$$Pv^k = \text{constant}, \quad k = \frac{c_p}{c_v} \tag{2.23}$$

or

$$\frac{P_1}{P_2} = \left(\frac{v_2}{v_1}\right)^k \tag{2.24}$$

Combining this with the equation of state for a perfect gas,

$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{k-1} \tag{2.25}$$

$$\frac{T_1}{T_2} = \left(\frac{P_1}{P_2}\right)^{(k-1)/k} \tag{2.26}$$

$$\frac{v_1}{v_2} = \left(\frac{P_2}{P_1}\right)^{1/k} \tag{2.27}$$

Polytropic Process

When a gas follows a reversible process that includes heat transfer, the process generally proceeds such that a plot of log P versus log v is a straight line, as shown in Fig. 2.1.

From this can be written [13]

$$\frac{d\ \ln P}{d\ \ln v} = -n$$

$$d\ \ln P + n d\ \ln v = 0$$

or

$$Pv^n = \text{constant} \tag{2.28}$$

where

$$n = \frac{\ln(P_2/P_1)}{\ln(v_1/v_2)} \tag{2.29}$$

With this, the following relations can be written:

$$\frac{P_2}{P_1} = \left(\frac{v_1}{v_2}\right)^n \tag{2.30}$$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(n-1)/n} = \left(\frac{v_1}{v_2}\right)^{n-1} \tag{2.31}$$

Adiabatic Versus Polytropic Process

An adiabatic process is a reversible constant entropy process for an ideal gas without heat transfer, following the relationship

$$Pv^k = \text{constant}$$

A polytropic process is a reversible process for an ideal gas with heat transfer, and variable entropy, following the relationship

$$Pv^n = \text{constant}$$

where

$n = 1$ for isothermal process (constant temperature)
$n = \infty$ for isometric process (constant volume)

All compressor processes fall between isentropic ($n = k$) and isometric ($n = \infty$).

Industry-accepted practice is to use adiabatic equations for single-stage and air compressors, while polytropic equations are generally used for all other situations.

Remember:

- $H_{ad} < H_p$
- $\eta_{ad} < \eta_p$
- $GHP_{ad} = GHP_p$
- Summation of individual-stage adiabatic head (H_{ad}) does not equal the overall compressor adiabatic head.
- The summation of individual-stage polytropic head equals overall compressor polytropic head.

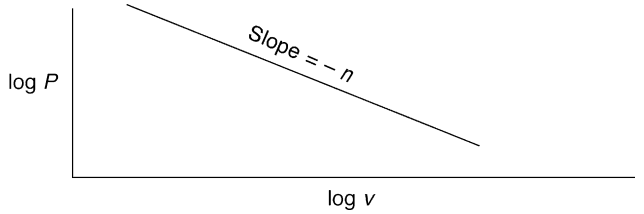


FIG. 2.1 Polytropic process. A plot of log P versus log v is a straight line. *(Data from Van Wylen G, Sonntag R (University of Michigan). Fundamentals of classical thermodynamics. New York: John Wiley and Sons; 1968.)*

HEAD

For a reversible adiabatic process [14],

$$Pv^k = \text{constant}$$

$$P_1 v_1^k = Pv^k = \text{constant}$$

$$v = \left(\frac{P_1}{P}\right)^{1/k} v_1$$

$$H_{\text{ad}} = \int_{P1}^{P_2} v\,dP$$

$$= \int_{P_1}^{P_2} (P_1/P)^{1/k} v_1\,dP$$

$$= \frac{k}{k-1} P_1 v_1 \left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right]$$

Using $Pv = ZRT$,

$$H_{\text{ad}} = ZRT_1 \frac{k}{k-1} \left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \tag{2.32}$$

For a polytropic process,

$$Pv_n = \text{constant}$$

$$\therefore H_p = ZRT_1 \frac{n}{n-1} \left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right]$$

$$n = \text{polytropic exponent}$$

$$n = \frac{\log r_p}{\log r_v} = \frac{\ln r_p}{\ln r_v} \tag{2.33}$$

$$r_v = \text{volume ratio} = v_1/v_2, \quad r_v > 1$$

$$r_p = P_2/P_1$$

$$\frac{n}{n-1} > 1$$

Mollier Method

If a Mollier chart of gas properties is available, the Mollier Method may be used [11]. Here is how the head equation for this method is derived.

$$H_p = ZRT_1 \frac{n}{n-1} \left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right] = ZR\frac{n}{n-1}(T_2 - T_1)$$

$$= (P_2 v_2 - P_1 v_1)^{(n-1)/n}$$

but

$$\left(\frac{P_2}{P_1}\right)^{(n-1)/n} = \frac{T_2}{T_1} = \frac{P_2 v_2}{P_1 v_1}$$

or

$$\frac{n-1}{n}=\frac{\ln\left(P_2v_2/P_1v_1\right)}{\ln(P_2/P_1)}$$

or

$$\frac{n}{n-1}=\frac{\ln\left(P_2/P_1\right)}{\ln(P_2v_2/P_1v_1)}$$

Therefore

$$H_p=\ \ln\left(\frac{P_2}{P_1}\right)\left[\frac{P_2v_2-P_1v_1}{\ln\left(P_2v_2/P_1v_1\right)}\right] \tag{2.34}$$

The last half of the equation is a log mean. Substituting an arithmetic mean introduces an insignificant error.

Therefore

$$H_p=\left[\ \ln\frac{P_2}{P_1}\right]\left(\frac{P_1v_1+P_2v_2}{2}\right) \tag{2.35}$$

This can be rearranged to

$$H_p=72\left[\ \ln\frac{P_2}{P_1}\right](P_1v_1+P_2v_2) \tag{2.36}$$

$$=166\left[\log\frac{P_2}{P_1}\right](p_1v_1+p_2v_2) \tag{2.37}$$

Adiabatic head can be determined from a Mollier diagram by using the relationship

$$H_{\mathrm{ad}}=h_{2_{\mathrm{ad}}}-h_1 \tag{2.38}$$

Note: $h_{2_{\mathrm{ad}}}$ is found by following a constant entropy line from point 1 to point 2 (see Fig. 2.2).

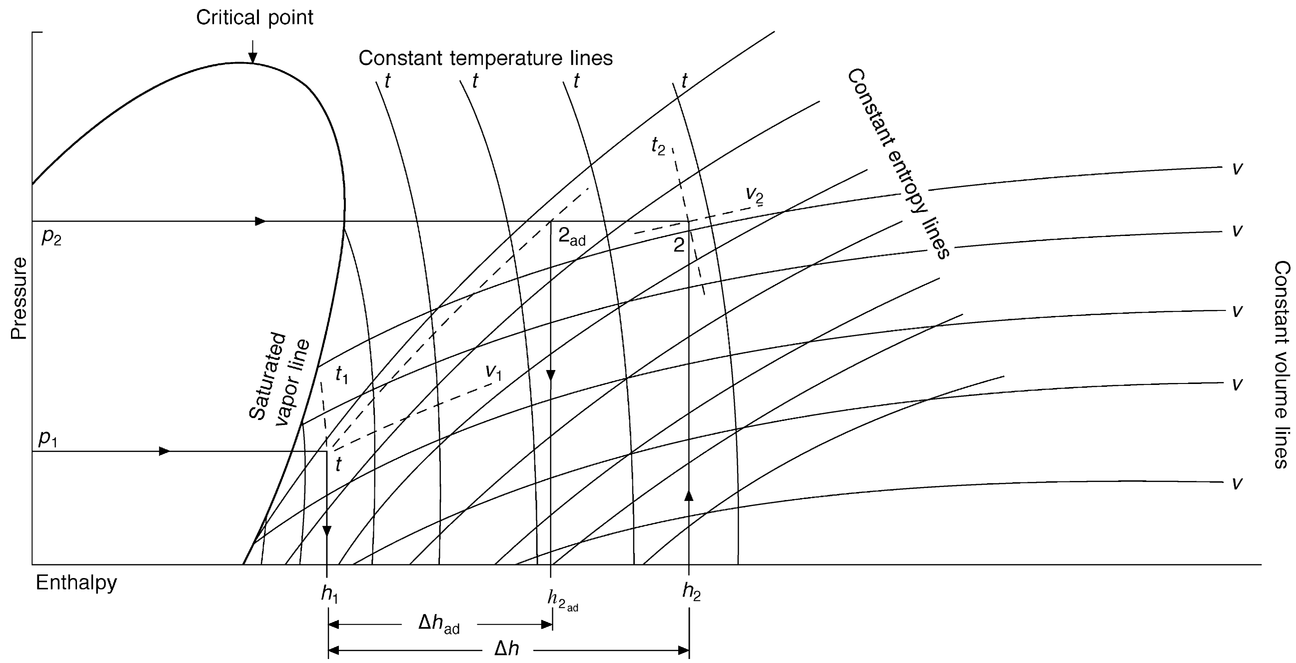


FIG. 2.2 Mollier diagram. *(With permission Compressor refresher. Jeannette, PA: Elliott Co.; 1975.)*

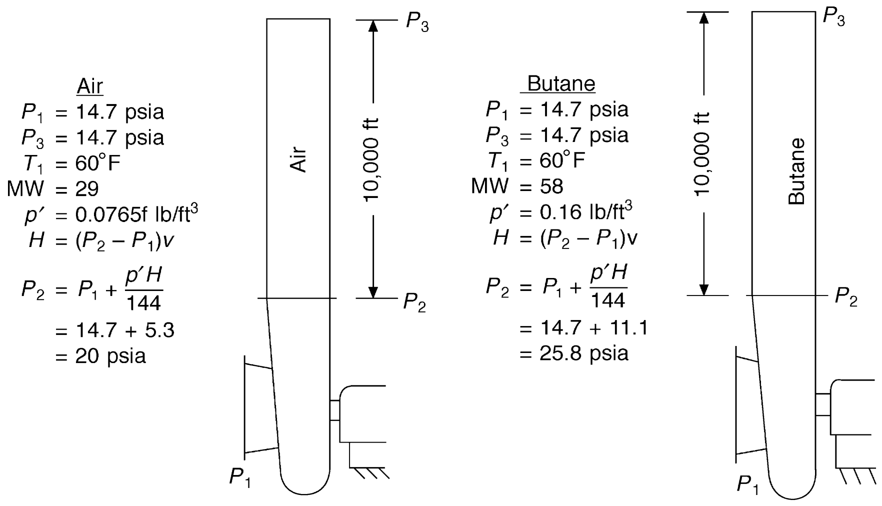


FIG. 2.3 Two identical compressors provide the same head but different discharge pressure due to a difference in gas density. *(Data from Paluselli DA. Basic aerodynamics of centrifugal compressors. Jeannette, PA: Elliott Co. Derrickson GW. Thermodynamic review. Jeannette, PA: Elliott Co.; 1968.)*

Conceptualizing Head

Head = Energy
= Enthalpy
= Foot-pound force per pound mass, or ft-lb_f/lb_m

For a constant speed only, CFM and head properly describe a particular compressor. Head is a function of speed and impeller geometry (diameter, tip opening, blade angle, etc.) and is not related to type of gas, temperature, or pressure. Discharge pressure for a given head is a function of gas mole weight, pressure, and temperature.

Imagine two incompressible gas columns, both 10,000 ft. high, both at atmospheric pressure and temperature at the uppermost ends, one column filled with air (MW = 29) and the other with butane vapor (MW = 58). For both cases, compressor geometry and speed are identical (Fig. 2.3). Since the gas properties are different, the discharge pressures for the two machines are significantly different, while the head (10,000 ft.) is the same.

This demonstrates that for a given head, a higher pressure ratio is obtained for a higher density gas.

WORK AND EFFICIENCY

The efficiency of a thermodynamic system is the ratio of work output of the system (head) to the work input to the system (shaft power).

The difference between head and work is the amount of losses internal to the machine due to such things as friction and windage. These losses show up as heat and add to the discharge temperature.

These losses include

External to the main flow path

Unavailable energy added

- windage and disk friction
- leakage

Internal to the main flow path

Actual losses of blade input energy:

- skin friction
- blade loading and diffusion
- incidence angle
- exit mixing losses
- clearance losses

Work

Using the general energy equation (first law of thermodynamics),

$$q + u_1 + P_1 v_1 + \frac{V_1^2}{2g} + Z_1 = W + u_2 + P_2 v_2 + \frac{V_2^2}{2g} + Z_2 \tag{2.15}$$

This can be reduced to

$$q + h_1 + \frac{V_1^2}{2g} = W + h_2 + \frac{V_2^2}{2g}$$

Assuming negligible heat transfer and minimal velocity effects,

$$W(\text{work}) = (h_2 - h_1) \tag{2.16}$$

For an isentropic process,

$$W = c_p(T_2 - T_1) \tag{2.39}$$

Adiabatic Efficiency

Adiabatic efficiency uses isentropic relationships to define head (useful work) and total work input.

$$\begin{aligned}\eta_{ad} &= \frac{H_{ad}}{W} \\ &= \frac{\Delta h_{\text{isentropic}}}{\Delta h_{\text{actual}}} \\ &= \frac{c_p(T_{2_{ad}} - T_1)}{c_p(T_2 - T_1)}\end{aligned} \tag{2.40}$$

where $T_{2_{ad}}$ represents isentropic compression (see Figs. 2.4 and 2.2).

$$\begin{aligned}\eta_{ad} &= \frac{T_{2_{ad}} - T_1}{T_2 - T_1} \\ \eta_{ad} &= \frac{\Delta T_{\text{isentropic}}}{\Delta T_{\text{actual}}} \\ \frac{T_{2ad}}{T_1} &= \left(\frac{P_2}{P_1}\right)^{(k-1)/k} \\ T_{2ad} - T_1 &= T_1\left(\frac{T_{2ad}}{T_1} - 1\right) = T_1\left[(r_p)^{(k-1)/k} - 1\right]\end{aligned} \tag{2.41}$$

This gives

$$\eta_{ad} = \frac{T_1\left[(P_2/P_1)^{(k-1)/k} - 1\right]}{T_2 - T_1} \tag{2.42}$$

Another form of the adiabatic efficiency equation [9]:

$$\eta_{ad} = \frac{RT_1 1\text{n}(P_2/P_1)}{c_p(T_2 - T_1)} \tag{2.43}$$

The overall adiabatic efficiency is useful as a measure of the overall performance of a turbocompressor for the purpose of determining power. However, it is not always a true indication of efficiency regarding internal losses. Fig. 2.4A illustrates this point. Since isentropic work is proportional to temperature rise ($W_{ad} = c_p \Delta T$), the distance from point 1 to point 2_{ad} is proportional to the adiabatic work required to compress the gas from P_1 to P_2. The actual work, however, is proportional to the vertical distance from point 1 to point 2.

For the three-stage compressor shown in Fig. 2.4C, note that this variation is further amplified. Also, note that the vertical distance (work) for the individual states W_a, W_b, plus W_c is greater than W_{ad} (total adiabatic work). This is due to the fanning effect of the pressure lines. $W_{ad} < (W_a + W_b + W_c) < W_p(1 - 2)$. Consider a compressor with an infinite

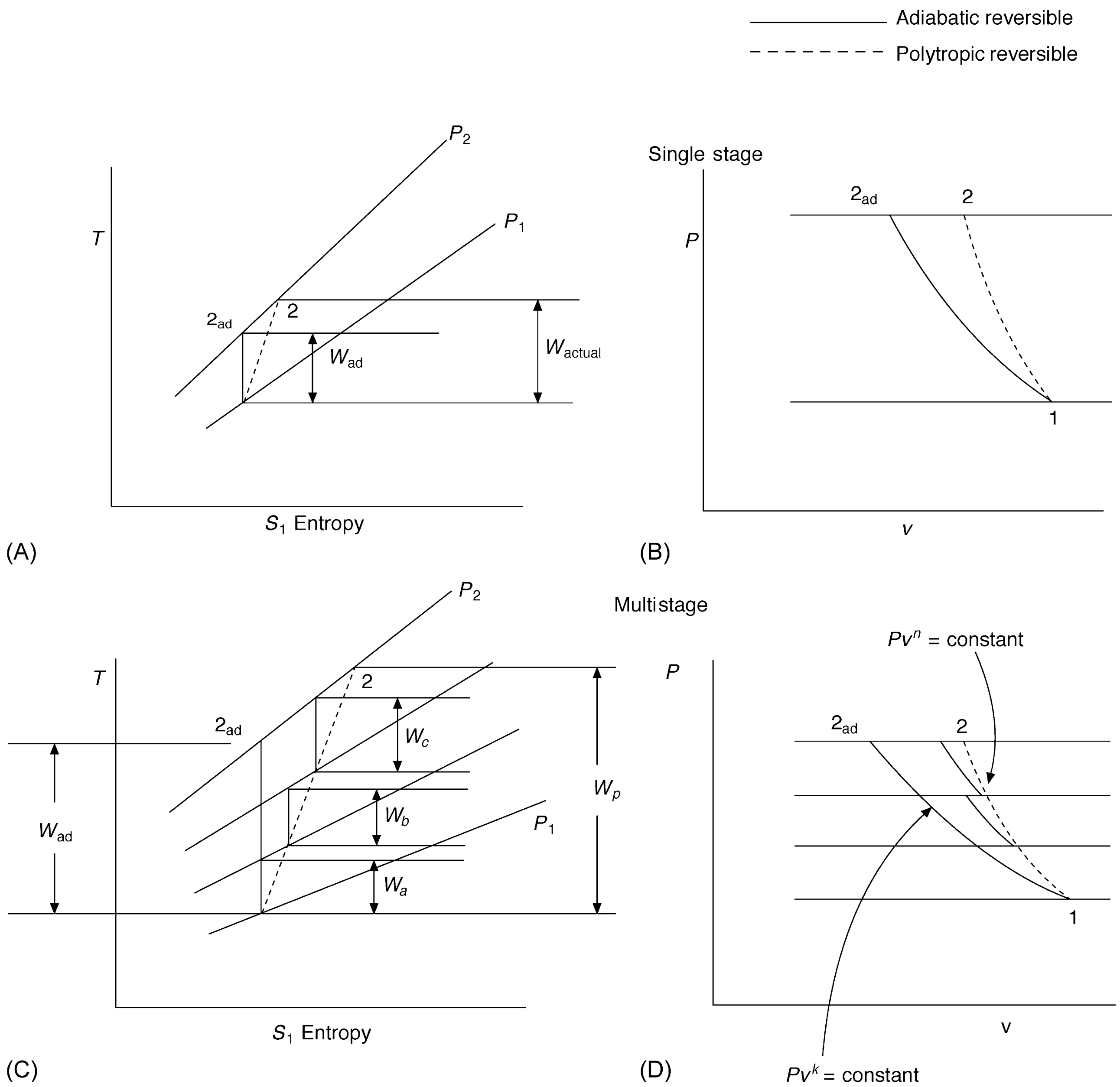


FIG. 2.4 Adiabatic versus polytropic process. The overall adiabatic efficiency is not always a true indication of efficiency regarding internal losses. Since isentropic work is proportional to temperature rise ($W_{ad} = c_p\Delta T$), the distance from point 1 to point 2_{ad} is proportional to the adiabatic work required to compress the gas from P_1 to P_2. The actual work, however, is proportional to the vertical distance from point 1 to point 2. For a three-stage compressor, this variation is further amplified. Note that the summation of individual-stage adiabatic head (H_{ad}) does not equal the overall compressor adiabatic head; however, the summation of individual-stage polytropic head equals overall compressor polytropic head.

number of stages. Then $W_{ad} < (W_1 + W_2 + W_3 + \cdots + W_\infty) = W_p(1-2)$. The following polytropic efficiency equation can be developed.

$$\eta_p = \frac{[(k-1)/k]\ \ln(P_1/P_2)}{\ln(T_1/T_2)} = \frac{k/(k-1)}{n/(n-1)} \tag{2.44}$$

This polytropic equation represents the true aerodynamic efficiency of a compressor for compression of an ideal gas [9]. There are, however, limitations to this equation also. Real gases do not always have a constant k value. The value of k for some gases at discharge conditions can vary significantly from the k value at suction conditions. The enthalpy (or Mollier) Eq. (2.45) is the most accurate means of calculating the aerodynamic efficiency for any condition as it does not depend on the k value being a constant. In some cases, it is the only equation that can provide accurate results. For this reason, real gas

equations of state like BWR equations as used in Gas Flex are the only means to calculate the head and efficiency when k is not constant. *Never* use an average value for k.

$$\begin{aligned}\eta_p &= \frac{\text{Head}}{\text{Work input}} \\ &= \frac{Hp}{h_2 - h_1}\end{aligned} \tag{2.45}$$

Horsepower

Horsepower is the rate of doing work.

$$\begin{aligned}\text{HP} &= \frac{\text{Work}}{\text{Time}} \\ \text{GHP} &= W \times \dot{M} \\ &= (h_2 - h_1)\dot{M} \\ &\text{Since}\end{aligned} \tag{2.46}$$

$$\eta = \frac{H}{W}, \quad W = \frac{H}{\eta} \tag{2.47}$$

Visualize a compressor lifting a weight of gas to a height (H) at a specified rate ($\dot{M}$).

$$\text{GHP} = \frac{H \times \dot{M}}{\eta} \tag{2.48}$$

FLOW MEASUREMENT, ORIFICE METERS

In order to derive the equation for flow through an orifice, we can use Bernoulli's equation with Station 1 being immediately upstream of the orifice and Station 2 at or immediately following the orifice [15].

$$\begin{aligned}P_1 v + \frac{V_1^2}{2g} &= P_2 v + \frac{V_2^2}{2g} \\ V_2^2 - V_1^2 &= 2gv(P_1 - P_2)\end{aligned}$$

For an incompressible fluid,

$$Q_{s_1} = Q_{s_2} = VA, \quad V^2 = Q_s^2/A^2$$

Substituting:

$$\begin{aligned}\frac{Q_s^2}{A_2^2} - \frac{Q_s^2}{A_1^2} &= 2gv\Delta P \\ Q_s &= \frac{A_1 A_2}{\sqrt{A_1^2 - A_2^2}}\sqrt{2gv\Delta P} \\ &= \frac{A_2}{\sqrt{1 - (A_2/A_1)^2}}\sqrt{2gv\Delta P} \\ A &= \frac{\pi d^2}{4} \Rightarrow \frac{A_2}{A_1} = \left(\frac{d}{D}\right)^2 \\ \beta &= \frac{d}{D} = \text{Beta Factor} \\ Q_s &= A_2 \frac{1}{\sqrt{1 - \beta^4}}\sqrt{2gv\Delta P} \\ Q_s &= \frac{\pi d^2}{4}\frac{1}{\sqrt{1 - \beta^4}}\sqrt{2gv\Delta P}\end{aligned}$$

where

d = orifice diameter, in.
D = pipe diameter, in.

$$\frac{1}{\sqrt{1-\beta^4}} = \text{Velocity of Approach Factor}, E \tag{2.49}$$

Ideally, d should be the diameter of the vena contracta, but this value is unknown, or rather cannot be directly measured. Instead, a correction or efficiency factor is used. This factor is best determined by calibration, but can be calculated (see Chapter 10). This factor is referred to as the discharge coefficient, C. Incorporating this into the flow equation,

$$Q_s = \frac{\pi d^2 CE}{4}\sqrt{2gv\Delta P} \tag{2.50}$$

C and E are sometimes combined and called the discharge coefficient, K.

$$Q_s = \frac{\pi d^2 K}{4}\sqrt{2gv\Delta P} \tag{2.51}$$

The ratio of the differential pressure, $\triangle P$, to the upstream static pressure, P, will influence the value of the overall flow coefficient. This occurs because there is a slight density change as the gas passes through the orifice meter. This effect is compensated for by the expansion factor, Y. The factor can be calculated or taken from a table or chart (see Fig. 10.4).

We now have the flow equation

$$Q_s = \frac{\pi d^2 KY}{4}\sqrt{2gv\Delta P} \tag{2.52}$$

Finally, to compensate for the thermal expansion of the orifice plate, the thermal expansion factor Fa is used. Generally, this factor can be assumed to be 1.00 at moderate temperatures unless extreme accuracy is necessary (see Fig. 10.3).

Including the thermal expansion effects,

$$Q_s = \frac{\pi d^2 KY\,\mathrm{Fa}}{4}\sqrt{2gv\Delta P} \tag{2.53}$$

GAS MIXTURES

One way to deal with gases that do not significantly vary from the ideal gas laws is described in the following procedures used for calculating k (specific heat ratio and adiabatic exponent) and the compressibility factor z (correction for deviation from ideal gas law, $Pv = ZRT$). This method has been around for quite some time and gives reasonable results. However, the best way to deal with any gas mixture, especially gases that significantly vary from the ideal gas laws, such as heavy hydrocarbons, more complex real gas equations of state such as Benedict, Webb & Rubin (BWR), Lee Kesler, or others should be used [35]. The software Gas Flex suggested at the back of this book and the additional reading section use BWR equations of state.

The properties of a gas mixture required for adiabatic compressor calculations are

1. Gas constant (dependent on molecular mass MW)
2. k, specific heat ratio and adiabatic exponent
3. P_1, T_1, v_1, and P_2
4. Compressibility, Z
5. Critical pressure, P_c
6. Critical temperature, T_c

Of these properties of a gas mixture, MW, c_p, c_v, P_c, and T_c are calculated by adding the products of the individual mol fractions of each constituent, times its specific property (see Table 2.1). The temperature of any constituent is obviously the temperature of the mixture. The v (specific volume) of the mixture is obtained from $Pv = ZRT$. The compressibility of a mixture is obtained from a compressibility chart using the calculated values of P_c and T_c of the mixture (see Eq. (2.45). Polytropic efficiency calculations for heavy hydrocarbon gases require determination of enthalpy).

The k value of a mixture is determined from

TABLE 2.1 Procedure for Calculating Values for a Gas Mixture following Perfect Gas Laws

Gas Mixture	(1)	(2)	(3)	(4)	(5)	(6)[a]	(7)[a]	(8)	(9)	(10)[a]	(11)
	Mol% of Each Gas	Mols/h of Each Gas	Mol Mass	(1) × (3)	Mass %	T_c	P_c	(1) × (6)	(1) × (7)	Mcp	(1) × (10)
......	......	......	......	*a*	$a/d \times 100$	......	......	......	......	......	......
......	......	......	......	*b*	$b/d \times 100$	......	......	......	......	......	......
......	......	......	......	*c*	$c/d \times 100$	......	......	......	......	......	......
				d				......	......		......
Calculate $k_{(\text{mixture})} = \dfrac{\sum Mcp_{(\text{mix})}}{\sum Mcp - 1.985}$				Apparent mol mass of mixture				$P_{c(\text{mix})}$	$T_{c(\text{mix})}$		ΣMcp

[a]*See Table A.1 for items 6, 7, and 10.*

$$\begin{aligned} Mc_p - Mc_v &= 1.985\,\text{BTU}/\left(\text{lb}-\text{mole}^\circ\text{R}\right) \\ &= 8.314\,\text{kj/kmol}^\circ\text{K} \\ k &= c_p/c_v \end{aligned}$$

or

$$\begin{aligned} k &= \frac{Mc_p}{Mc_v} \text{ where } M = \text{molecular weight of the gas} \\ &= \frac{Mc_p}{Mc_p - 1.985} \text{ where } Mc_p = \text{MW} \times c_p, \text{or molal } c_p \end{aligned}$$

For a gas mixture, the summation of the mole fraction times the molal c_p of each constituent is used (Table 2.1).

$$k = \frac{\Sigma Mc_p(\text{mix})}{\Sigma Mc_p(\text{mix}) - 1.985} \tag{2.54a}$$

or for metric values

$$k = \frac{\Sigma Mc_p(\text{mix})}{\Sigma Mc_p(\text{mix}) - 8.32} \tag{2.54b}$$

The compressibility (Z_1) of the mixture can be determined by finding the reduced temperature T_{R1} and the reduced pressure P_{R1} (using Table A.1) as follows

$$T_{R1} = \frac{T_1}{T_{c(\text{mix})}} \tag{2.55}$$

$$P_{R1} = \frac{P_1}{P_{c(\text{mix})}} \tag{2.56}$$

Then these values are entered in Fig. A.1 (in Appendix A) to find Z.

THERMODYNAMIC STATE EQUATIONS

Compressor performance cannot be accurately predicted without detailed knowledge of the behavior of the gas or gases involved.

Thermodynamic state equations are developed from experimental data or derived from kinetic theory or statistical mechanics. The thermal equation of state relates state variables, usually, temperature, T, pressure, P, and density, ρ.

The caloric equation of state relates the energy content of the fluid to state variables, usually in terms of specific internal energy, e, or specific enthalpy, h [35].

Thermally Perfect Gas

The most commonly used thermal equation of state is the perfect gas equation, $P = \rho RT$ (Eq. (2.5)). Its limitation for real fluids is best realized by the fact that all gases can be liquefied. The highest temperature at which liquid and vapor can coexist defines the critical point for the fluid. From observed critical point properties (T_c, P_c, ρ_c), it is known that all gases are far from thermally perfect at this point. Use of the thermally perfect gas model should be limited to temperatures much higher than T_c and pressures much less than p_c to insure reasonable accuracy. Basic thermodynamics shows that the energy content of a thermally perfect fluid is a function of temperature only, i.e., $h^0 = h^0(T)$ (for real gas models, the superscript, 0, is used to designate parameters for thermally perfect gases) [35].

Real Gases

The general form of the thermal equation of state for real gases is $Pv = ZRT$ (Eq. (2.7)) where the gas compressibility factor, Z may be obtained from established data (Fig. A.1) or from an appropriate real gas equation of state. A highly accurate real gas model is the eight-parameter Benedict-Webb-Rubn (BWR) equation used in Gas Flex:

$$P = \rho RT + \left(B_0 RT - A_0 - C_0/T^2\right)\rho^2 + (bRT - a)\rho^3 + a\alpha\rho^6 + \frac{c\rho^3}{T^2}\left(1 + \gamma\rho^2\right)e^{-\gamma\rho^2} \tag{2.57}$$

where A_0, B_0, C_0, a, b, c, α, γ are constants for the gas or gas mixture.

For real gases, h is a function of pressure as well as temperature. The effects of pressure are expressed as differences with respect to the perfect gas state (h^0) and referred to as departure functions [35].

The specific heat is calculated by using curve fit coefficients.

$$c_p = A + BT + CT^2 + DT^3 + ET^4 + FT^5 \tag{2.58}$$

where A, B, C, D, E, & F are constants for the gas or gas mixture.

By combining the coefficients based on the gas mixture, gas mixture coefficients are established and used in the various equations to determine the gas mixture properties.

k Values

For real gases, there are three different k values. The three k values are equal for ideal gases; however, the values will diverge for real gases.

1. The first k value is the ratio of specific heats (c_p/c_v). This value varies only as a function of temperature. There is no pressure correction.
2. The second k value is the isentropic temperature exponent. k is calculated by the expression

$$\left(\frac{P_1}{P_2}\right)^{\left(\frac{k-1}{k}\right)} = \frac{T_1}{T_2}.$$

This value is normally used for flow element calculations. N method compressor head calculations also use this k value.

The third k value is the isentropic volume exponent. k is calculated by the expression

$$\left(\frac{P_1}{P_2}\right) = \left(\frac{v_2}{v_1}\right)^k$$

where v is the specific volume calculated by the expression $v = ZRT$. This value is used for sonic velocity calculations.

The behavior of a wide variety of gases—in any conceivable mixture—can be accurately computed using real gas equations. [35]. Enthalpy and specific volume can be determined directly if the pressure and temperature of the gas or gas mixture are known. This enables very accurate calculation of head, efficiency, and flow.

Mollier Diagrams

Compressor performance cannot be accurately predicted without detailed knowledge of the behavior of the gas or gases involved.

Mollier diagrams are readily available for most pure gases at "conventional" pressures and temperatures. However, in cryogenic areas or at very high pressure, some gas behaviors are difficult to predict. Gas properties in these areas therefore have been estimates determined through rather empirical methods.

The behavior of a wide variety of gases—in any conceivable mixture—can be accurately computed, plotted, and offered to the process engineer in the form of a Mollier diagram [11].

The only input required to obtain a plot of gas behavior is the identity and proportion of the gases involved (if a gas mix), and the limiting pressure and temperature values.

Temperature entropy (T-S) and pressure enthalpy (P-h) curves are commonly used to display gas properties. While these curves are primarily used for steam or refrigeration cycles, the curves are very useful for any process. The process is easy to "visualize" when plotted on a Mollier diagram. One can "see" the phase change, the expansion, or the compression process, and therefore easily comprehend the overall process and the effect of process changes.

Steam Generation

A very common cycle depicted in Fig. 2.5 is the Rankine cycle of the steam-generating plant.

Refrigeration

The reverse of the Rankine or Carnot cycle is the vapor compression or refrigeration cycle (Fig. 2.6).

Economizer cycles are commonly used to improve system efficiency (Fig. 2.7).

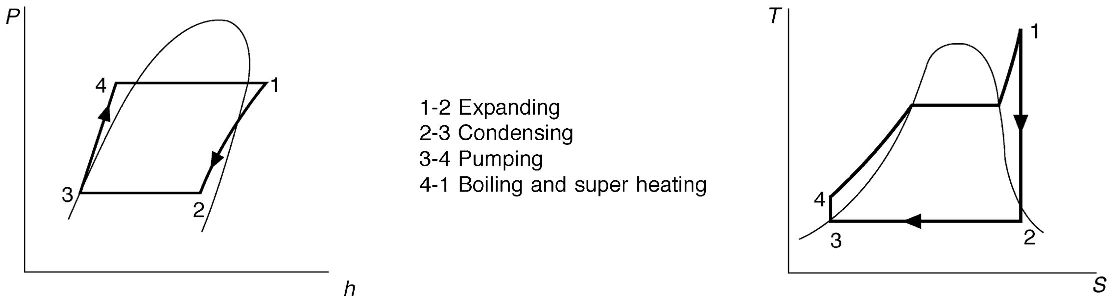


FIG. 2.5 Rankine cycle. *(With permission Compressor refresher. Jeannette, PA: Elliott Co.; 1975.)*

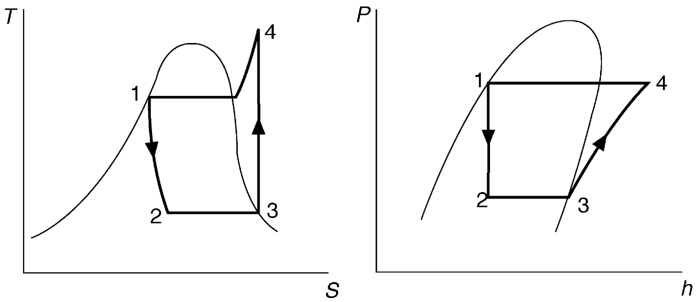


FIG. 2.6 Carnot cycle. *(With permission Compressor refresher. Jeannette, PA: Elliott Co.; 1975.)*

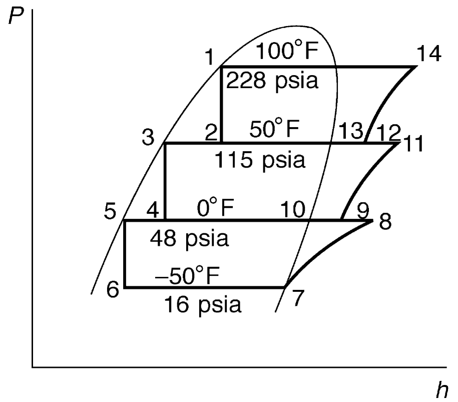


FIG. 2.7 Economizer cycle. *(With permission Compressor refresher. Jeannette, PA: Elliott Co.; 1975.)*

Refrigeration is second only to air compression as the most common application of centrifugal compressors.

In the early years of the twentieth century, most mechanical refrigeration capacity was used to produce manufactured ice for the preservation of food. These units used gases such as ammonia and carbon dioxide. Gradually, the scope was increased until today refrigeration is applied in many areas other than the home, commercial, and comfort fields for which the fluoridated hydrocarbons are used extensively.

Refrigeration is used in such processes as the manufacture of ethylene, ammonia, alkylation, and dewaxing. Most industrial refrigeration is an area associated with oil refining and therefore uses the readily available hydrocarbons as refrigerants.

The use of hydrocarbons as refrigerants has advantages other than availability. They have high mol weights, which permit fewer compressor impellers to reach condensing pressure. Also, there is a sufficient variety of hydrocarbons to provide a relatively good choice of refrigerants in the temperature range from –300°F to +60°F.

The most commonly used hydrocarbon refrigerants are:

Gas	Cooling Temperature Range
Methane	–300° to –200°F
Ethylene and ethane	–175° to –75°F
Propylene and propane	–50° to 0°F
Butane	+10° to +60°F

Refrigeration capacity refers to the total amount of heat absorbed in the cooler. This is generally referred to in tons of refrigeration or more simply Tons.

The unit of Ton came about from the early days of cooling, in which ice was used in almost all cases. The natural amount of "cold" or "coldness" was a given weight of ice. A meaningful amount was a Ton. Therefore, the definition of a ton of refrigeration, or just a Ton, is the amount of heat absorbed by a ton of ice at 32°F melting to a ton of water at 32°F in a period of 24 h. Since the latent heat of fusion of ice is 144 BTU/lb, a Ton, then, is equivalent to the absorption of 288,000 BTU/day or 12,000 BTU/h or 200 BTU/min.

PERFORMANCE COEFFICIENTS

In order to condense the large number of variables in the study and design of aerodynamic components, dimensionless coefficients are used extensively. The coefficients are developed via Buckingham's Pi theorem, where repeating variables are systematically combined to form dimensionless groups. Some of the useful dimensionless groups plus some other frequently used values follow.

Reynolds number

$$\begin{aligned} \text{Rey\#} &= \frac{\text{Inertial Forces}}{\text{Viscous Forces}} = \frac{\text{Momentum}}{\text{Friction}} \\ &= \frac{VD}{v'} \end{aligned} \tag{2.57}$$

$$\frac{VD\rho}{\mu'} \tag{2.58}$$

Mach number

$$\begin{aligned} \text{Mach\#} &= \frac{\text{Inertial Forces}}{\text{Compressibility Forces}} = \frac{\text{Momentum}}{\text{Elasticity}} \\ &= \frac{V}{a} \end{aligned} \tag{2.59}$$

Flow coefficient

$$\frac{700}{Nd^3}Q = \varphi \tag{2.60}$$

Head coefficient

$$\frac{H}{U^2/g} = \mu \tag{2.61}$$

Work coefficient

$$\frac{W}{U^2/g} = \gamma \tag{2.62}$$

Efficiency

$$\eta = \frac{\mu}{\gamma} \tag{2.63}$$

where

Q = volume flow CFM
N = speed, RPM
d = impeller dia. in.
H = head ft-lb_f/lb_m
U = tip speed, FPS

$$U = \frac{\pi dN}{720} = \frac{dN}{229.18}$$

g = gravitational constant 32.2 ft-lb_m/lb_f-s^2

$$W = \text{work} = h_2 - h_1 \text{ ft-lb}_f/\text{lb}_m$$

Equivalent tip speed, FPS

$$U\sqrt{\theta}$$

where

$$\sqrt{\theta} = \sqrt{\frac{26.2\,\text{MW}}{kTZ}} = \frac{\text{Speed of Sound — Air}}{\text{Speed of Sound — Gas}} \tag{2.64}$$

Specific speed

$$N_s = \frac{N\sqrt{Q/60}}{H^{0.75}} \tag{2.65}$$

Fan Laws

From these relationships, off-design performance can be approximated. These relationships are most accurate for single-stage compressors and an ideal gas. For multistage compressors with gas that deviates from the ideal gas laws, the accuracy of the following equations is reduced. Eqs. (2.66), (2.67) are graphically represented in Fig. 2.8.

$$Q \propto N \tag{2.66}$$

$$H \propto N^2 \tag{2.67}$$

$$\ln r_p \propto N^2 \tag{2.68}$$

$$(T_2 - T_1) \propto N^2 \tag{2.69}$$

$$\mathrm{GHP} \propto N^3 \tag{2.70}$$

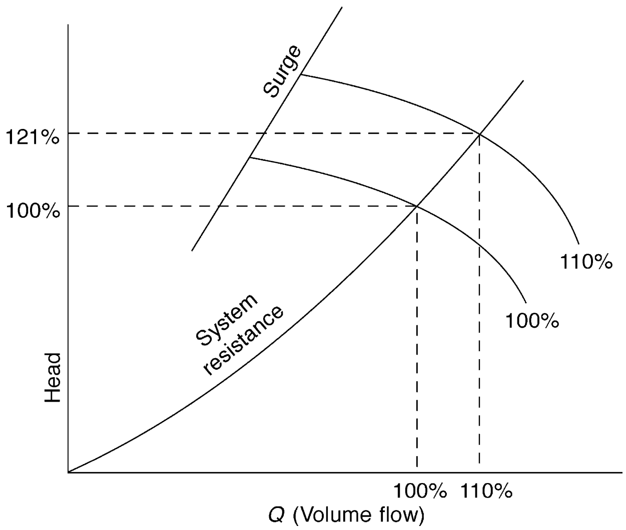


FIG. 2.8 Fan laws. *(Data from Paluselli DA. Basic aerodynamics of centrifugal compressors. Jeannette, PA: Elliott Co. Compressor refresher. Jeannette, PA: Elliott Co.; 1975.)*

Chapter 3

Aerodynamic Components

The design of the primary components of both axial and centrifugal compressors is relatively simple. Complexities arise in trying to get everything to work together with a high degree of efficiency—mechanically as well as aerodynamically. This chapter describes those main components, explains their functions, and covers some areas that need special consideration.

AXIAL COMPRESSORS

The major components of an axial compressor consist of (a) inlet nozzle, (b) prewhirl vanes, (c) rotating vanes, (d) stator vanes, (e) dewhirl vanes, and (f) discharge nozzle (Figs. 3.1 and 3.2).

The nozzle guides and accelerates the gas stream into the prewhirl vanes, which turn the gas stream to properly align it with the rotating blades. While both the rotating and stationary vanes act as diffusers, the rotating vane's primary function is to add to the total energy of the gas stream. Stator vanes, acting both as diffusers and reversing blades, orient the flow properly for the next row of rotating blades. While the first few rows of stator vanes are generally adjustable to compensate for off-design conditions, most rows of stator vanes are fixed. The dewhirl vanes remove the swirl from the gas stream before it enters the diffuser section.

CENTRIFUGAL COMPRESSORS[1]

The major elements of the centrifugal compressor (Fig. 3.3) consist of (a) the inlet nozzle, (b) inlet guide vanes, (c) impeller, (d) radial diffuser, (e) return channel, (f) collector volute, and (g) discharge nozzle.

The inlet nozzle accelerates the gas stream and directs it into the inlet guide vanes, which may be fixed or adjustable.

On a multistage compressor, the inlet nozzle is generally radial. In this case, the inlet guide vanes are necessary to properly distribute the flow evenly to the first-stage impeller (Fig. 3.4). Single-stage compressors frequently incorporate an axial inlet. In this case, inlet vanes may not be necessary.

Simply stated, kinetic energy is imparted to the gas via the impeller by centrifugal forces. The diffuser then reduces the velocity and converts the kinetic energy to pressure energy (Fig. 3.5).

Because of the rotational effects of the impeller, the gas travels through the diffuser in a spiral manner. Therefore, before entering the next impeller, the flow must be straightened out by the return channel vanes (Fig. 3.6).

Diaphragms

A diaphragm consists of a stationary element, which forms half of the diffuser wall of the former stage, part of the return bend, the return channel, and half of the diffuser wall of the later stage. Due to the pressure rise generated, the diaphragm (Fig. 3.7) is a structural as well as an aerodynamic device.

For the last stage or for a single-stage compressor, the flow leaving the diffuser enters the discharge volute. It is common to design these volutes for constant angular momentum ($R_i V_i =$ constant). This generally results in some velocity change through the volute (V_2 to V_5). Once the gas leaves the volute, it passes through the discharge nozzle, which reduces the velocity somewhat before entering the process piping. Fig. 3.8 represents such a constant angular momentum volute.

Since velocities are relatively high through the diffuser section (several hundred feet per second), surface finish/friction factor is crucial to overall efficiency of the unit.

1. Adapted from "Compressor components," M. Sassos, Elliott Company, Jeannette, PA, 1986, with permission [16].

Compressor Performance. https://doi.org/10.1016/B978-0-12-814219-6.00003-9

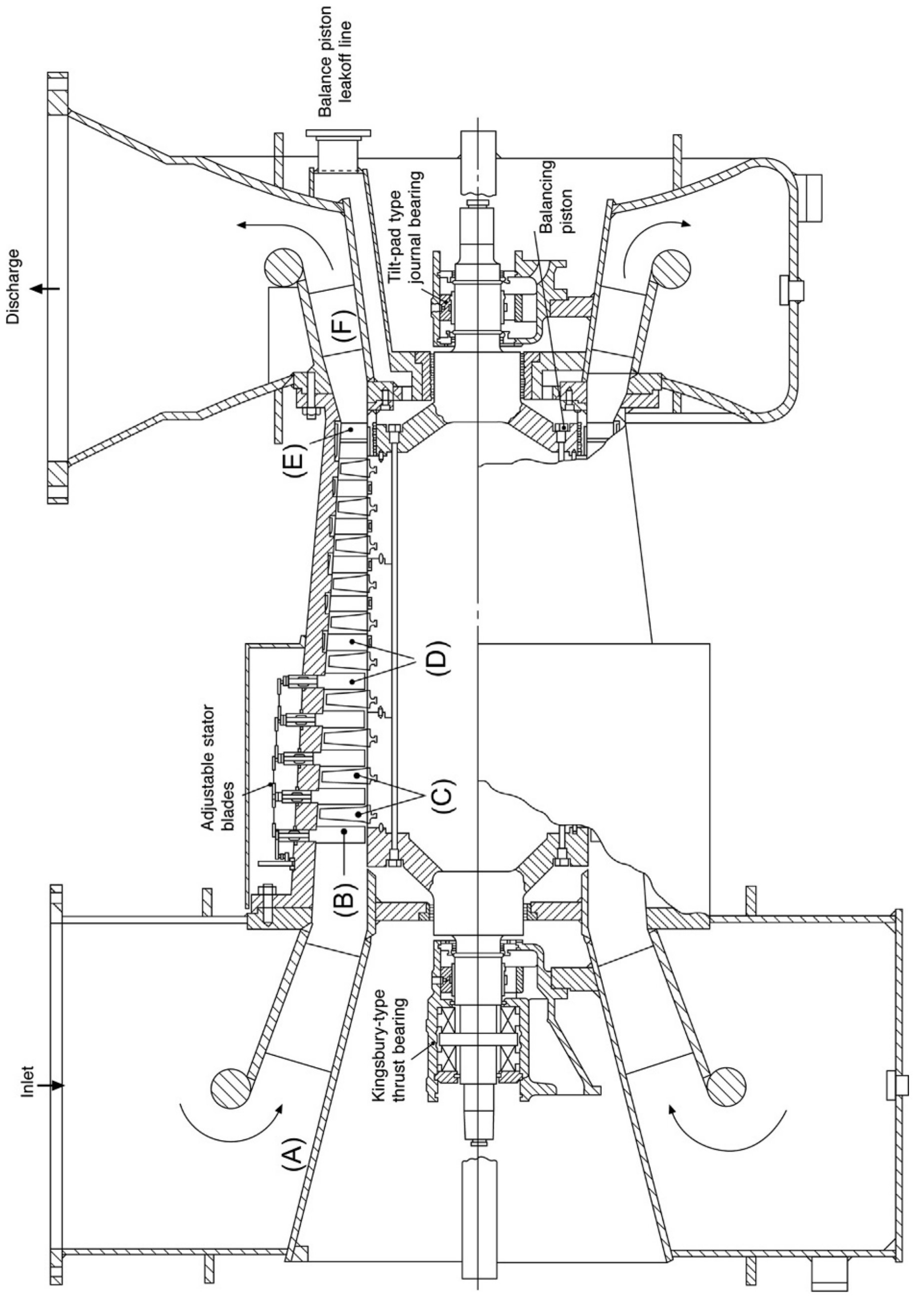


FIG. 3.1 Axial compressor. *(Courtesy of Elliott Company, with permission.)*

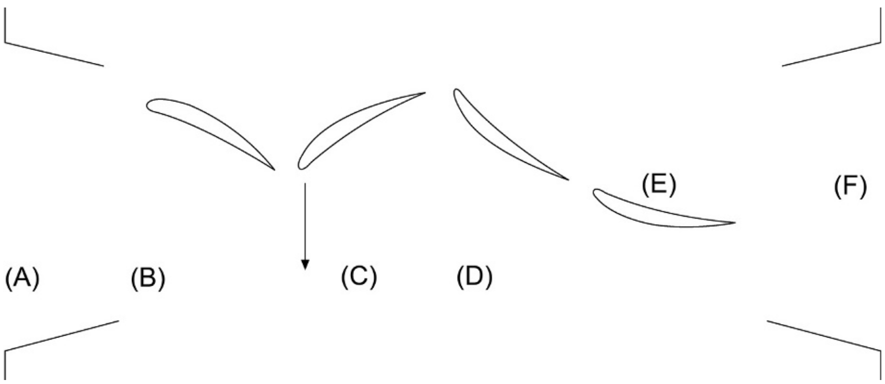


FIG. 3.2 Axial components: (a) inlet nozzle, (b) prewhirl vanes, (c) rotating vanes, (d) stator vanes, (e) dewhirl vanes, and (f) discharge nozzle.

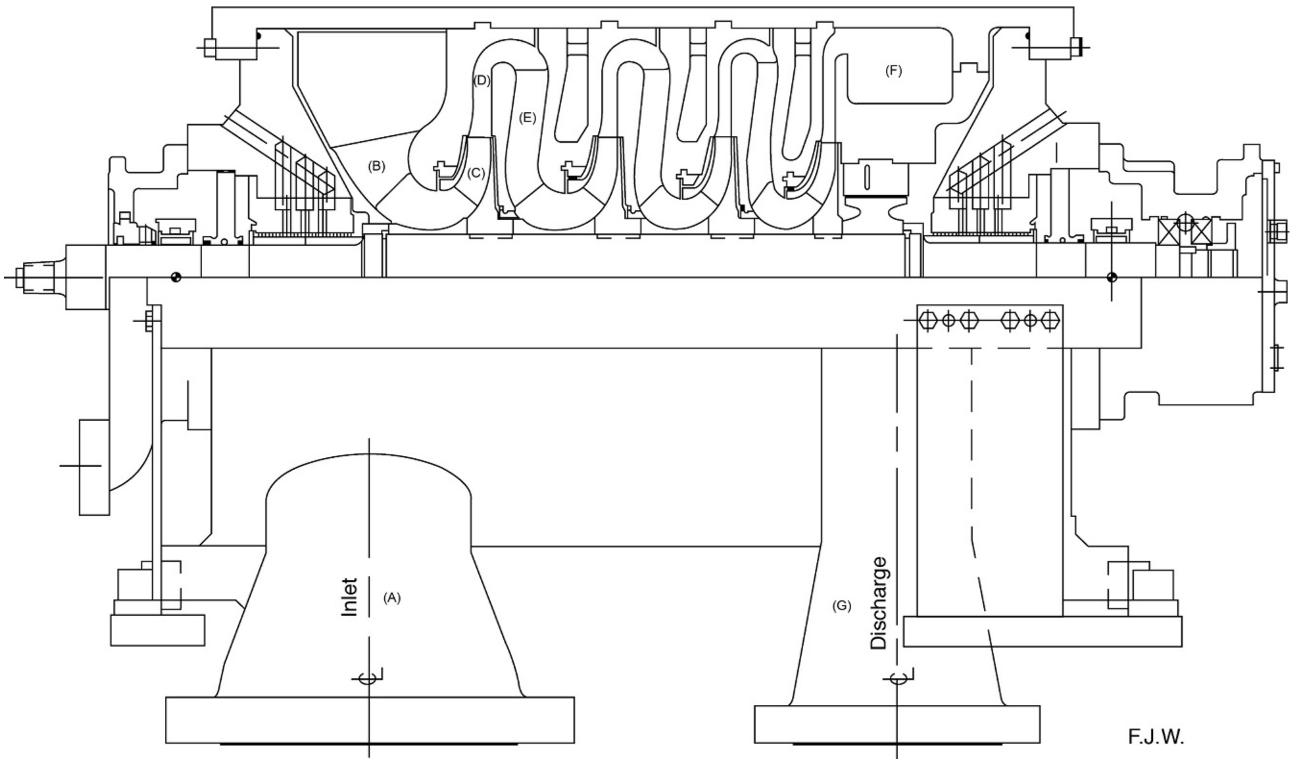


FIG. 3.3 Multistage centrifugal compressor: (a) the inlet nozzle, (b) inlet guide vanes, (c) impeller, (d) radial diffuser, (e) return channel, (f) collector volute, and (g) discharge nozzle.

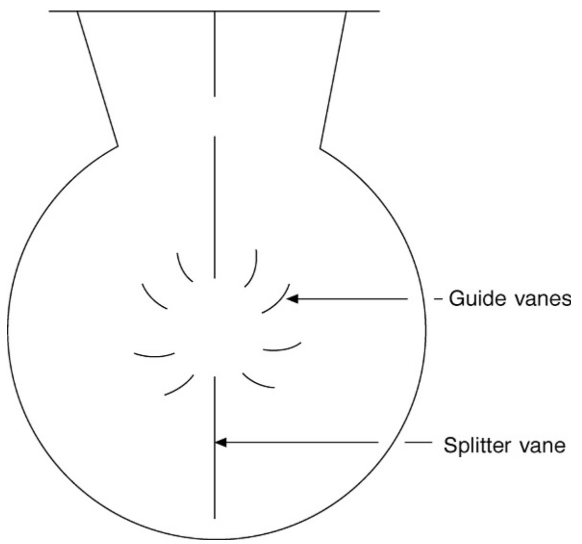


FIG. 3.4 Multistage compressor inlet showing splitter vanes and guide vanes.

In many processes, dirt or polymer buildup on the diaphragm surfaces will give the aerodynamic surfaces a rough finish (Fig. 3.9). In some cases, polymer buildup has been known to severely restrict the diffuser passage. Both conditions cause increased pressure losses and result in reduced overall efficiency of the compressor.

Surface finish in these critical areas can be enhanced and preserved by applying a nonstick coating such as fluorocarbon-based material, or a corrosion-resistant coating such as electroless nickel. The long-term effects are shown in Fig. 3.10.

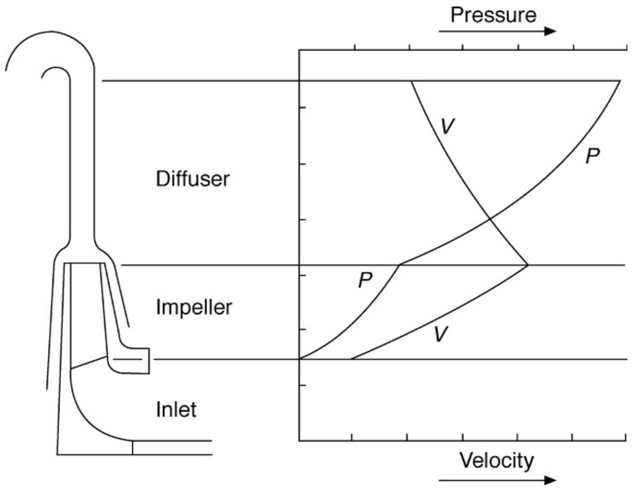


FIG. 3.5 Velocity/pressure development. *(Data from Paluselli DA. Basic aerodynamics of centrifugal compressors. Jeannette, PA: Elliott Co.)*

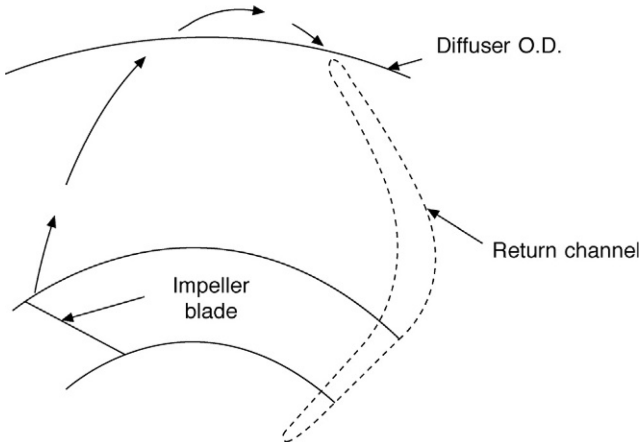


FIG. 3.6 Flow path of gas from impeller tip to return channel.

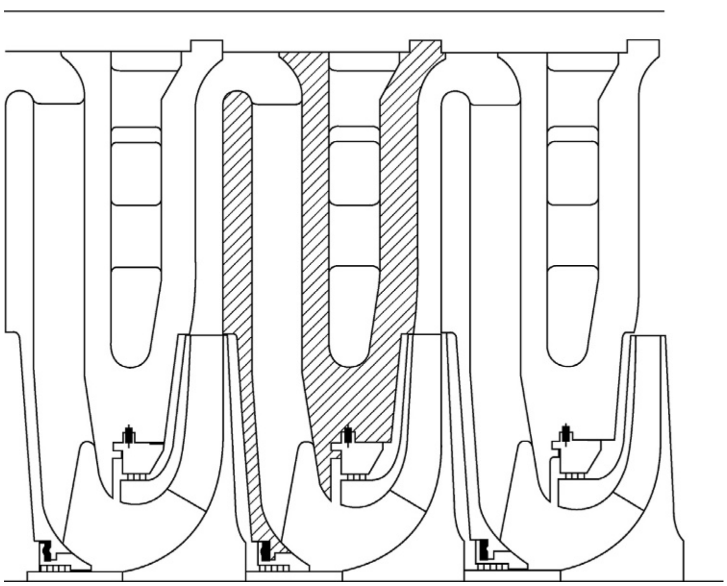


FIG. 3.7 Multistage centrifugal compressor diaphragm.

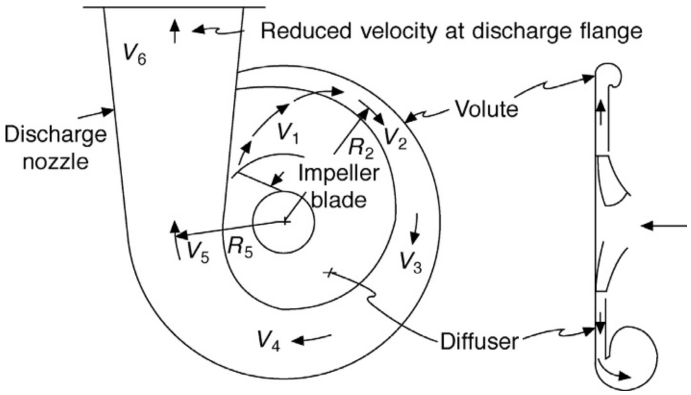


FIG. 3.8 Discharge volute.

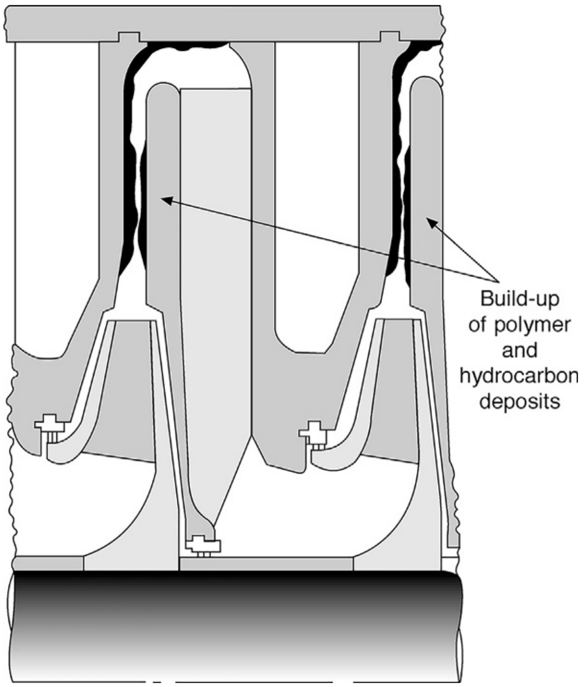


FIG. 3.9 Dirt or polymer buildup in diffuser passages of a centrifugal compressor.

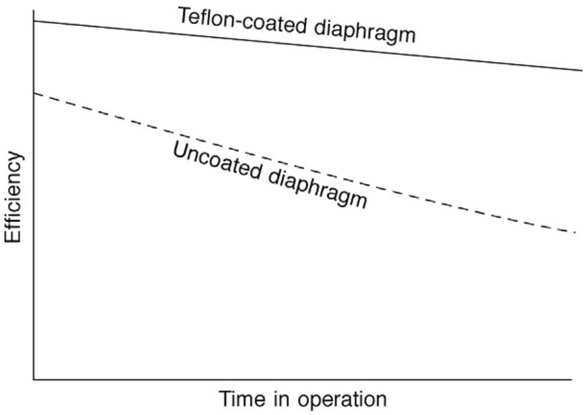


FIG. 3.10 Effect of coated and noncoated surfaces on dirt/polymer buildup in diffuser passages of a centrifugal compressor.

INTERSTAGE SEALS

Due to the pressure rise across successive compression stages, seals are required at the impeller eye and rotor shaft to prevent gas backflow from the discharge to inlet end of the casing. The condition of these seals therefore directly affects the compressor performance.

The simplest and most economical of all shaft seals is the straight labyrinth shown in Fig. 3.11. This seal is commonly utilized between compression stages and consists of a series of thin strips or fins, which are normally part of a stationary assembly mounted in the diaphragms. A close clearance is maintained between the rotor and the tip of the fins.

The labyrinth seal is equivalent to a series of orifices. Minimizing the size of the openings is the most effective way of reducing the gas flow. Labyrinths clogged with dirt (Fig. 3.12) and worn or wiped labyrinths with increased clearances (Fig. 3.13) allow larger gas leakage. This can affect compressor operation, and therefore the seals should be replaced.

Labyrinth material has typically been aluminum, because aluminum is compatible with most gases and is ductile enough to prevent rotor damage in the event of rubbing. A hard labyrinth material could result in dry whirl and catastrophic failure of the compressor.

Calculations and field performance data indicate that wiped interstage seals can decrease unit efficiency by 7% or more. Operating modes that contribute to labyrinth damage include extended surging, prolonged running in the critical speed regions, and liquid slugging.

In order to reduce or negate the performance effects common with damaged interstage seals, several improvements have been adopted by compressor manufacturers in the past several years. Most noteworthy is the use of abradable seals in the impeller eye and shaft seal areas. Advantages include tighter design operating clearances and minimal efficiency effects after a seal rub, as shown in Fig. 3.14.

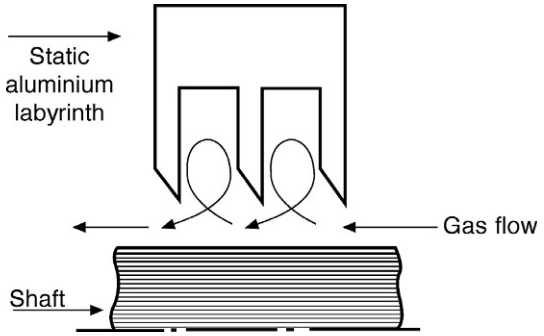


FIG. 3.11 Aluminum labyrinth. New and clean. Turbulence creates resistance to leakage flow.

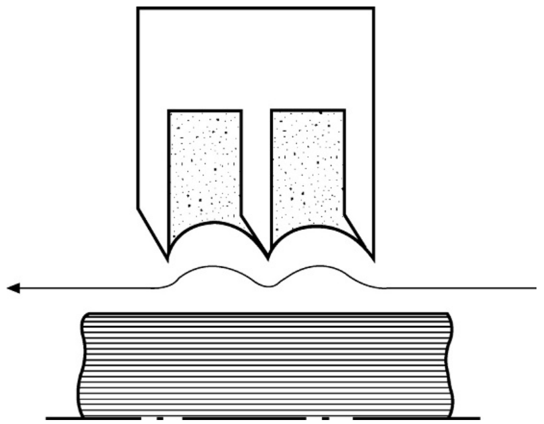


FIG. 3.12 Fouled labyrinth. Turbulence reduced, leakage flow increased.

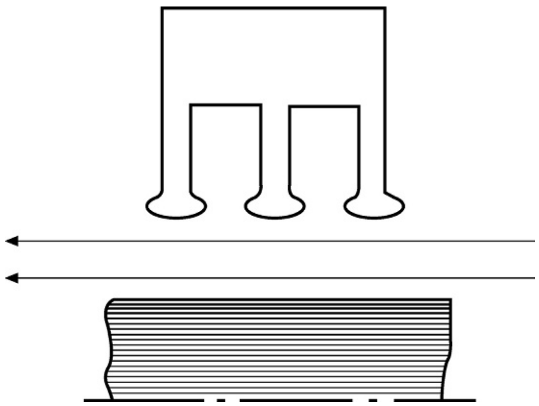


FIG. 3.13 Rubbed labyrinth. Clearance increased, turbulence reduced, leakage increased.

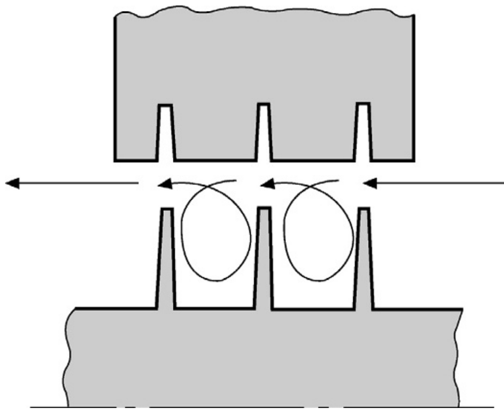


FIG. 3.14 Rubbed abradable seal. Running clearance unaltered, turbulence continues to create resistance to leakage flow.

The efficiency gain of abradable seals is achieved through the reduction of seal clearances, thereby reducing recirculating flow through the impellers. Impeller eye seals, interstage shaft seals, and balance piston seals are effective in improving compressor efficiency when changed to the abradable design. Abradable seals also control impeller thrust, which varies with seal clearance (Fig. 3.17).

Having the fins as the rotating element permits centrifugal force to prevent the buildup of process deposits. Where conventional static labyrinths are used on a fouling duty, buildup of deposits adversely affects the flow characteristic across the labyrinth, with detrimental effect on compressor efficiency. Rotating fins minimize this problem. See Figs. 3.11 and 3.12.

A rub on an aluminum labyrinth causes the tips of the aluminum fins to mushroom out (Fig. 3.13). This creates undesirable flow characteristics across the labyrinth and increases the radial clearance. Both these two factors are detrimental to compressor efficiency and will have an effect on the thrust loading of the machine. With the abradable design, the rotating fins rub into the static element, without damage to the fins and without effect on the normal running clearances. No performance deterioration or change in thrust load occurs (Figs. 3.14 and 3.15).

The overall efficiency improvement attainable by using abradable seals in a compressor varies with several factors, most notably the size of the compressor. Flow capacity increases as the square of the impeller diameter, while seal clearance increases more linearly with impeller size and is also dependent on other factors such as bearing clearances and manufacturing tolerances. Therefore, as the compressor size increases, the leakages involved become a smaller portion of the total flow. As this happens, the improvements gained by reducing these leakages have a diminishing impact on the machine's overall efficiency. Therefore, it is the smaller, higher pressure compressors that benefit most from abradable seal conversions.

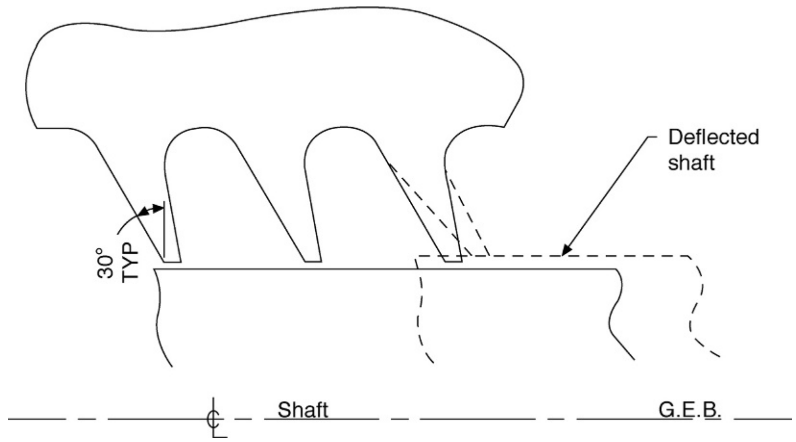


FIG. 3.15 Rub-tolerant polymer labyrinth seal. This type of seal made from Arlon, Torlon, or other thermoplastics is a realistic alternative to abradable seals. The rub-tolerant seal can be designed with abradable seal clearances and a flexible tooth design. This feature accommodates vibration excursions with negligible tooth wear. An additional benefit is the ability of the polymer material to withstand very corrosive environments.

BALANCE PISTON SEAL

A balance piston (or a center seal) is utilized to compensate for aerodynamic thrust forces imposed on the rotor due to the pressure rise through a compressor. The purpose of the balance piston is to utilize the readily available pressure differentials to oppose and balance most of these thrust forces. This enables the selection of a smaller thrust bearing, which results in lower horsepower losses.

A certain amount of leakage occurs across the balance piston since a labyrinth seal is utilized. This leakage, which is a parasitic flow (a horsepower loss), is normally routed back to the compressor suction, thus creating a known differential pressure across the balance piston (Fig. 3.16). Occasionally, leakoff may be routed to other sections to gain an efficiency advantage. Air compressors generally route the balance piston leakage to atmosphere.

Since the balance piston seal must seal the full compressor pressure rise, integrity of this seal is crucial to good performance. A damaged seal results in higher leakage rates, higher horsepower consumptions, and greater thrust loads.

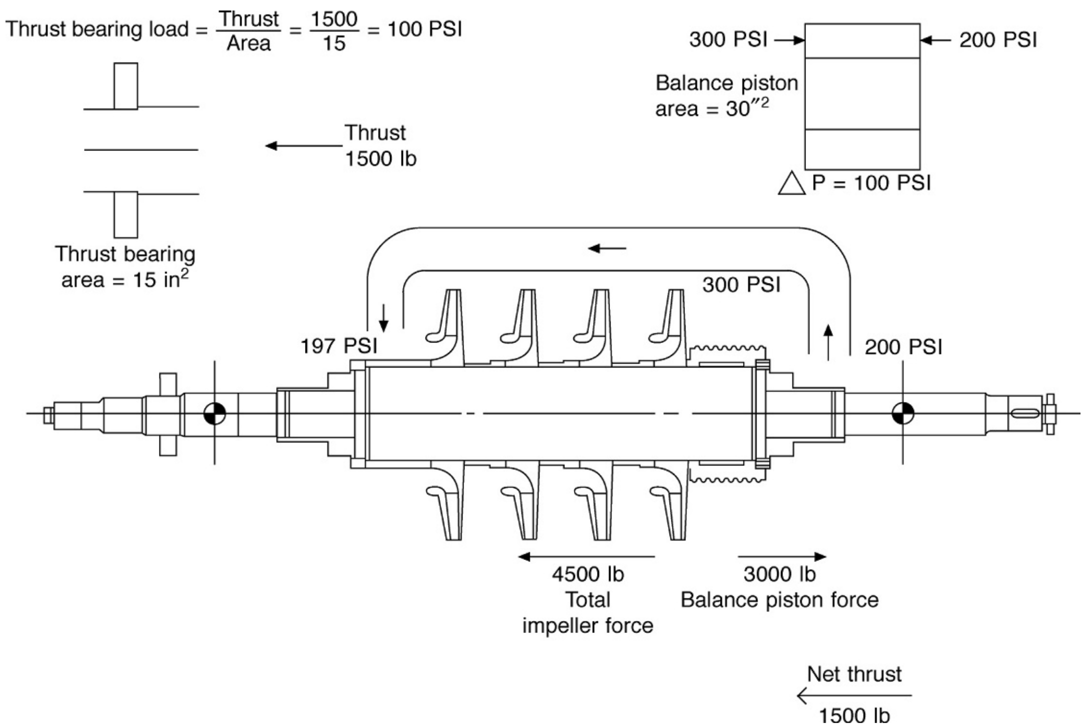


FIG. 3.16 Schematic of compressor thrust. Pressure drop in balance line is normally 1 to 3 PSI. *(Data from Compressor refresher. Jeannette, PA: Elliott Co.; 1975.)*

One user of a compressor noted the following data before and after a balance piston seal replacement. This machine was in refrigeration service and was required to maintain a constant discharge pressure.

		Before	After
Discharge pressure	(PSIG)	410	410
Discharge temperature	(°F)	142	116
Axial Position	(Mils)	24	19
Balance line ΔP	(PSID)	4.7	1.5
Speed	(RPM)	11,440	10,770
Thrust metal temperature	(°F)	240+	165

The balance piston damage was a result of surging and vibration excursions. The interstage seals were also extensively damaged, which contributed to the poor compressor efficiency. Note the differences in the various data for before and after the seal replacement. The discharge temperature was high, since more work input was required to achieve the desired discharge pressure. In order to get the higher level of work input, the speed was increased. The wiped seals not only caused increased inefficiencies, but also higher thrust loads. This showed up in the axial position and thrust bearing temperature.

IMPELLER THRUST[2]

Impeller thrust is generated by the differential force on the cover and hub of the wheel. These forces are the summation of the product of the pressures acting on the cover, hub, and the differential area from the shaft to the tip of the wheel.

The impeller generates thrust between the eye and tip of the wheel, as well as below the eye. These forces (thrust) are caused by several different effects:

1. rotational inertia field
2. leakage
3. friction
4. diffusion
5. momentum

The effects of friction and diffusion are secondary.

As indicated by the flow paths (Fig. 3.17), gas will flow toward the tip of the wheel along the hub, and toward the eye of the wheel along the cover. Due to the pressure rise in the diffuser, the return channel pressure is greater than the pressure behind the impeller hub. Leakage therefore occurs from the return channel toward the impeller hub and outward toward the impeller tip. The effect of the pressure established by this leakage, superimposed upon the rotating inertia field, is shown in Fig. 3.17A.

From Fig. 3.17, it can be seen that there is an obvious net pressure differential toward the suction of the machine in addition to the area caused by the eye of the impeller. This is indicated in Fig. 3.17B. Integrating the products of the pressure and area from eye to tip results in the net thrust on a wheel.

For a "perfect" seal at the impeller eye and shaft areas, the thrust is only a function of the area inside the eye seal. For a "real" seal with clearance and leakage in these areas, the net thrust is approximately 50% greater. As the clearances increase beyond the design values, thrust values increase even further.

EFFICIENCY IMPROVEMENTS

Increased competitive pressures have resulted in continued improvements to compressor design. Design tools not available 20 or 30 years ago, such as computational fluid dynamics (CFD), are now routinely used for analysis of compressor aerodynamic design.

Fig. 3.18 is a good example of modern high-efficiency staging design. Every feature has been designed with high efficiency in mind. Full, gentle, radiused flow paths designed with a meanline program like CompAero then further analyzed via CFD (Figs. 3.19 and 3.20) assure precise flow path control by minimizing gas velocity changes and flow separation for

2. Adapted from "Compressor refresher," Elliott Company, Jeannette, PA, 1975, with permission [11].

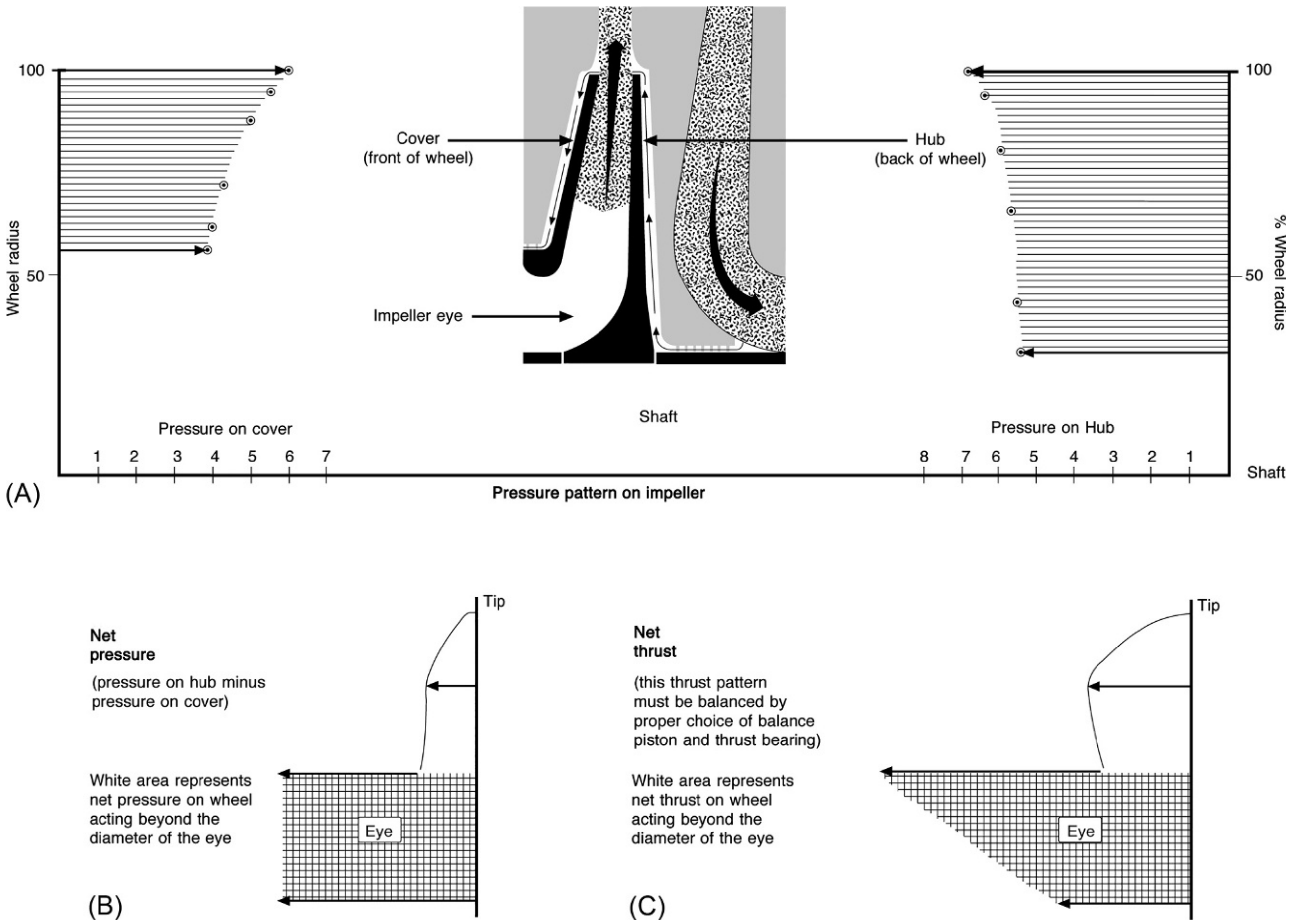


FIG. 3.17 (A) Pressure pattern on impeller. (B) Net pressure. (C) Net thrust. *(With permission Compressor refresher. Jeannette, PA: Elliott Co.; 1975.)*

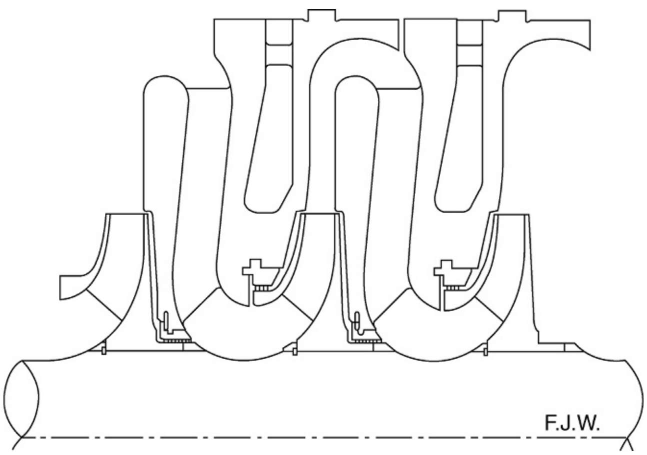


FIG. 3.18 High-efficiency centrifugal compressor stage.

the compressor stage as well as for the auxiliary flow paths such as the inlet and discharge nozzles and volute design. Abradable or rub-tolerant thermoplastic interstage shaft and impeller eye seals minimize recirculation of gas.

By including high-efficiency staging as shown in Fig. 3.18, abradable or rub-tolerant interstage and balance piston seals, and coated rotors and stationary hardware in a new or rebuilt compressor, overall flange-to-flange polytropic efficiencies of 85% or higher are achievable.

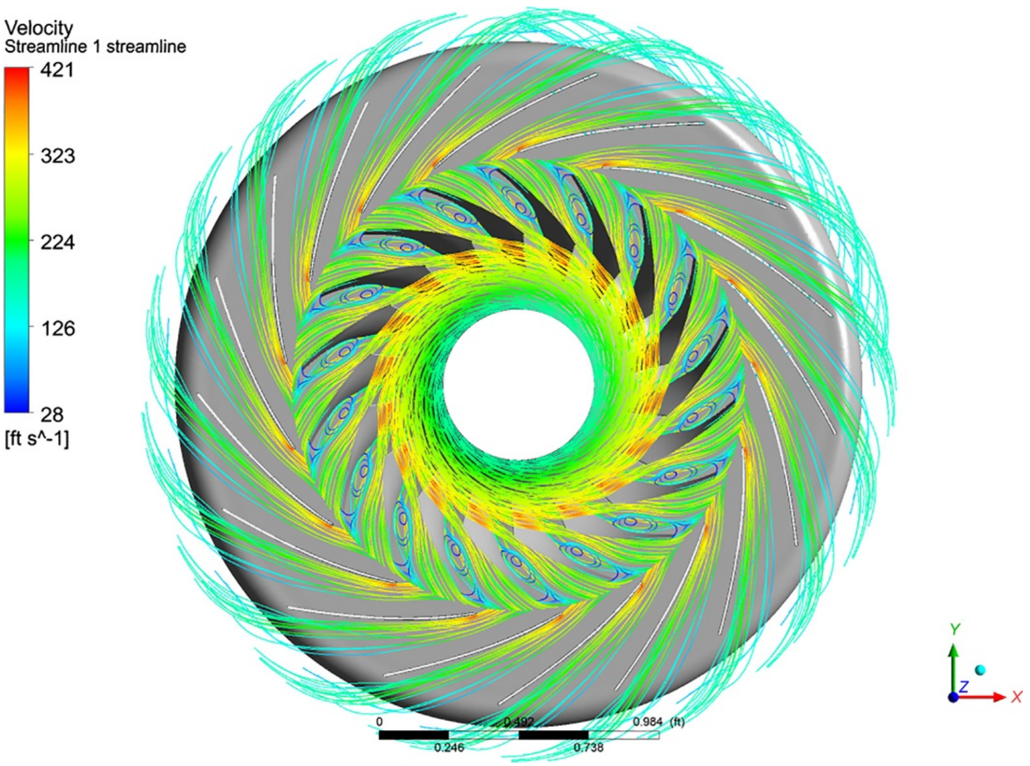


FIG. 3.19 CFD results showing streamlines for the preliminary design of a closed impeller with semi-inducer impeller blades and diffuser vanes. Note the flow separation area in the impeller. Further development of the impeller blade design to improve the impeller performance is still necessary. *(Courtesy of General Engineering Solutions, Gdynia, Poland.)*

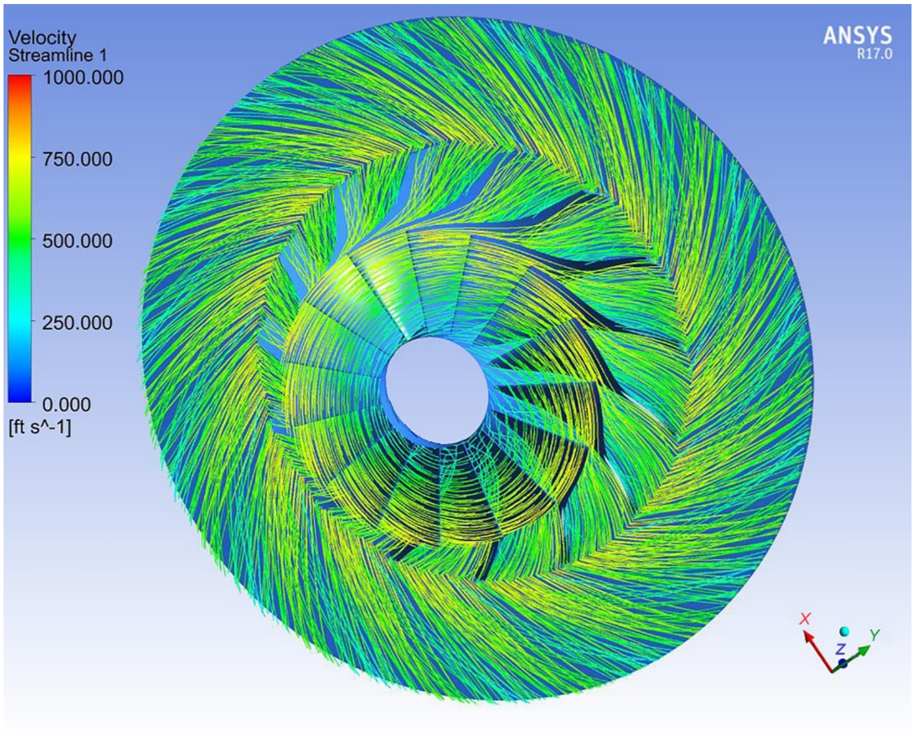


FIG. 3.20 CFD results showing streamlines for the final design of an open-wheel centrifugal impeller with full inducer blades. The impeller tip speed is 871 ft/s. The machine Mach number is 1.1. The flow coefficient is 0.15 (ref. Equation 2.60 in Chapter 2). The aerodynamic impeller efficiency is 0.936. *(Courtesy of Mechanical Solutions, Inc. Whippany, NJ, USA.)*

This page intentionally left blank

Chapter 4

Compressor Characteristics

As noted in Chapter 1, compressor performance is a function of the type of compressor. As well as the hardware configuration, the type of gas plays its part in determining the compressor characteristic curve. The effects of various parameters such as diffuser width, blade angle, gas density, compressor speed, and Mach Number in the development of the compressor characteristic curve shape are discussed in this chapter.

CENTRIFUGAL COMPRESSORS[1]

The characteristics of a centrifugal compressor (Fig. 4.1) are determined by the impeller, diffuser, and return channel or volute geometry.

There are three important aspects of the compressor curve that will be discussed (Fig. 4.2):

1. Slope of the curve
2. Stonewall (or choke)
3. Surge

SLOPE

To understand about the slope of the centrifugal compressor head curve, it is necessary to first understand what is going on at the impeller discharge in terms of velocity vector diagrams [18].

V_{rel} (Fig. 4.3) represents the gas velocity relative to the blade. U_2 represents the absolute tip speed of the blade. The resultant of these two velocity vectors is represented by V, which is the absolute velocity of the gas ($U_2 + V_{rel} = V$). Knowing the magnitude and direction of this absolute velocity, we can break this vector into its radial and tangential components (Fig. 4.4).

For a radial inlet impeller the head output is proportional to the product of U_2 and V_T.

For a typical backward-leaning bladed impeller, as the flow decreases at constant speed, V_{rel} decreases. This makes V_T increase, which increases head output. This head increase with decreasing flow is what causes the basic slope to the centrifugal compressor performance curve (Fig. 4.5).

Fig. 4.6 shows characteristic curves for three basic configurations: forward-leaning, radial, and backward-leaning blade profiles. Note that the forward-leaning blades provide a positive sloping head curve and the maximum head output. This is because V_T is increasing with increasing flow.

A radial bladed impeller has a theoretical constant (flat) head curve since V_T does not change with flow.

Overall stage efficiency is highest for backward-leaning impellers, while efficiency is lowest for forward-leaning blades.

For best efficiency, most modern centrifugal compressors use backward-leaning bladed impellers. Directionally speaking, the greater the backward lean, the better the efficiency. However, as the angle increases, the head is reduced (see Figs. 4.6C and 4.7). A designer can select a blade angle and tip width to best fit the desired head and efficiency characteristics of the particular application.

1. Adapted from “Centrifugal compressors … the cause of the curve,” D.C. Hallock, Elliott Company, Jeannette, PA, 1968, with permission [17]; and “Compressor performance,” R. Salisbury, Elliott Company, Jeannette, PA, with permission [18].

Compressor Performance. https://doi.org/10.1016/B978-0-12-814219-6.00004-0

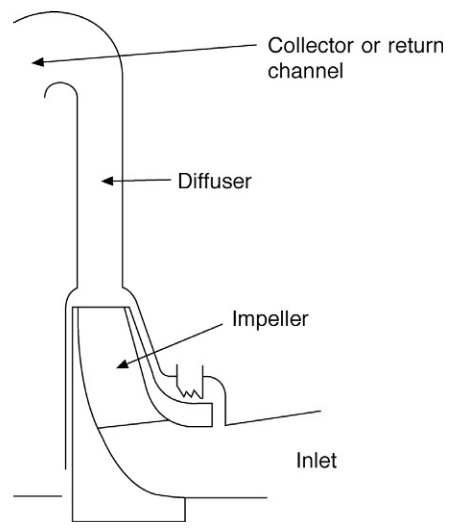


FIG. 4.1 Centrifugal compressor stage [18].

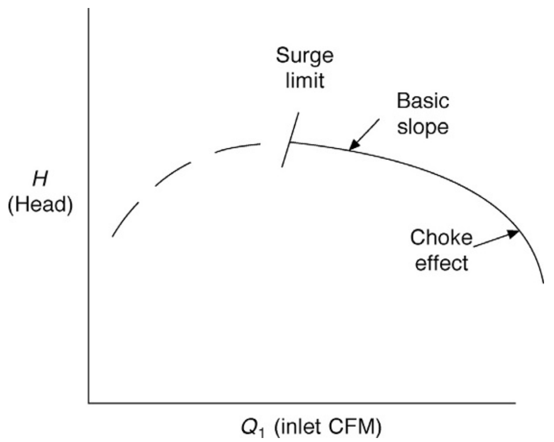


FIG. 4.2 Head curve for a compressor stage [17]. *(Used courtesy of Elliott Company, Jeannette, PA.)*

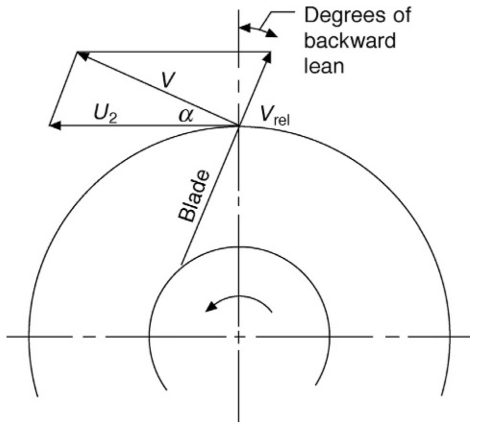


FIG. 4.3 Vector diagram of the gas velocity relative to the impeller blade. The slope of the characteristic curve is strongly influenced by this relationship [17]. *(Used with permission of Elliott Company, Jeannette, PA.)*

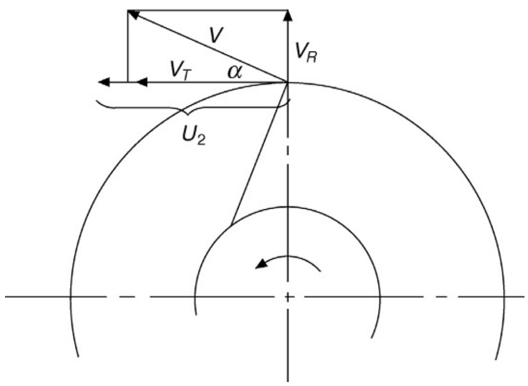


FIG. 4.4 Vector diagram of the gas velocity shown in Fig. 4.3 in radial and tangential components [17]. *(Used with permission of Elliott Company, Jeannette, PA.)*

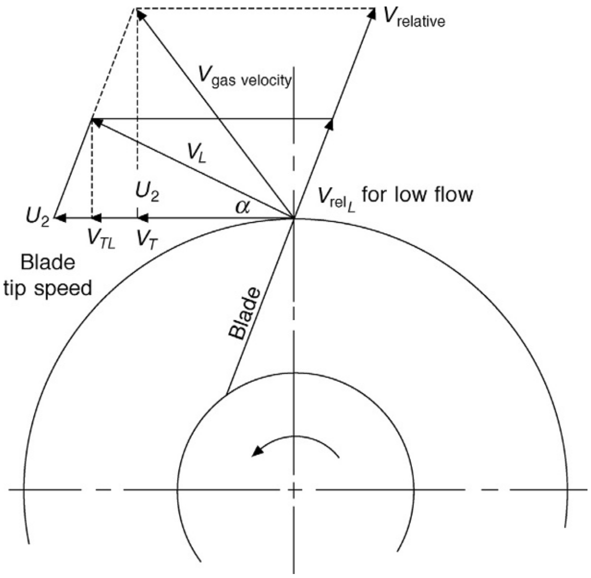


FIG. 4.5 The effect of a change in flow rate on the vector diagram at the impeller O.D. is shown. Note that V_T decreases as flow increases (V_{rel} increases) for a backward-leaning impeller blade. This gives the backward-leaning impeller the characteristic negative-sloping head curve shown in Fig. 4.2. *(Used with permission of Elliott Company, Jeannette, PA.)*

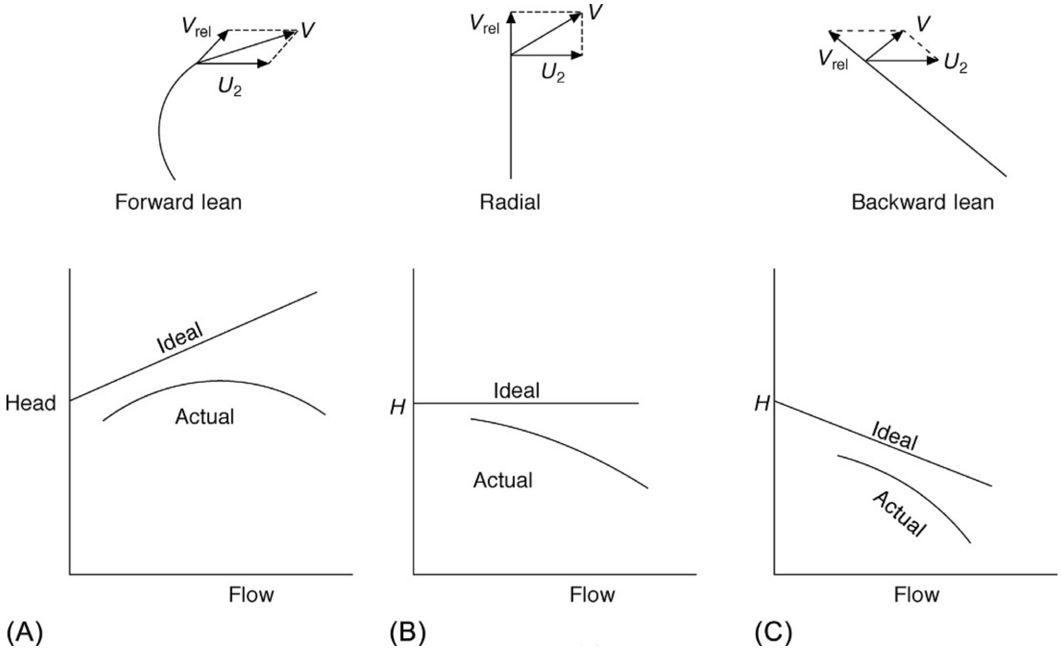


FIG. 4.6 Three basic head curve shapes for centrifugal compressors. *(Adapted from Sheperd DG (Cornell University). Principles of turbomachinery. New York: Macmillan; 1956, p. 67.)*

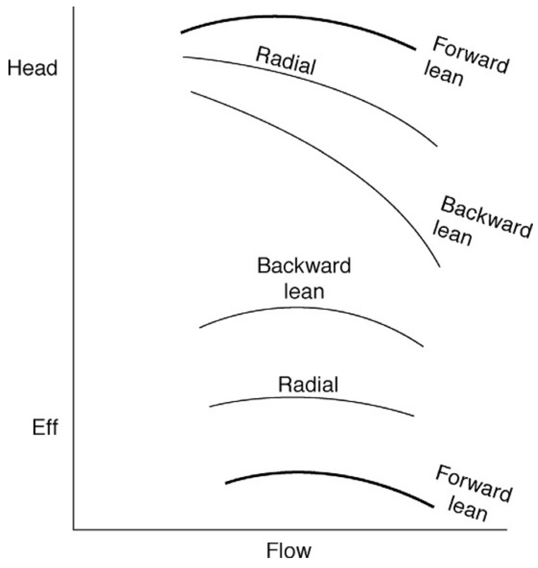


FIG. 4.7 The effect of impeller blade angle on head.

STONEWALL

Stonewall, or choke, is a condition at which increased capacity (flow) results in an excessive decrease in head (see Fig. 4.9). This occurs because the Mach number is approaching 1.0.

Operating at a very high-flow rate has very negative effects on the performance of a centrifugal compressor, and can sometimes be damaging (see Figs. 4.10 and 4.11). For axial compressors, high-flow rates can also create blade flutter and result in serious damage to the blading. The stonewall effect of the centrifugal compressor stage with a vaneless diffuser is controlled by impeller inlet vector geometry.

U_1 in Fig. 4.8 represents the tangential velocity of the leading edge of the blade. V represents the absolute velocity of the inlet gas, which, having made a 90° turn, is now moving essentially radially (in the absence of prewhirl vanes)—hence the name radial inlet. By vector analysis, V_{rel}, which is gas velocity relative to the blade, is of the magnitude and direction shown.

$$V = U_1 + V_{rel}$$

At design flow, V_{rel} lines up with the blade angles. As flow increases beyond design, V increases. As V increases so does V_{rel}. V_{rel} now impinges at a negative angle to the blade, a condition known as negative angle of attack. High negative angles of attack contribute to the stonewall phenomenon because of boundary-layer separation and a reduction of effective area in the blade pack. This area reduction, in addition to the already high V_{rel}, brings on Mach 1 and a corresponding shock wave, as shown in Fig. 4.9.

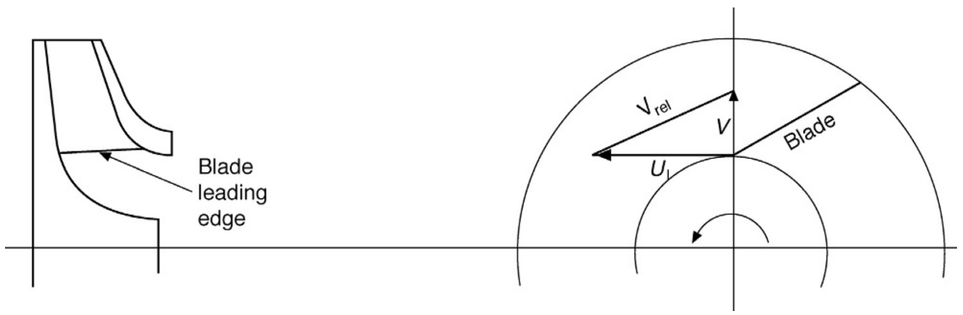


FIG. 4.8 Stonewall [17]. *(Used with permission of Elliott Company, Jeannette, PA.)*

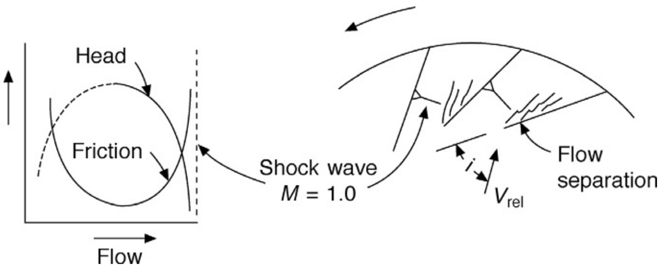


FIG. 4.9 Stonewall.

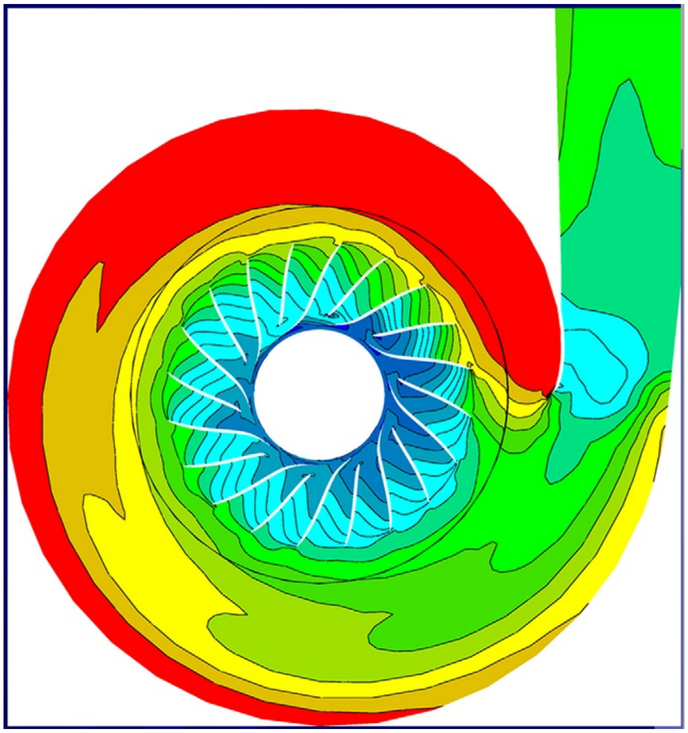


FIG. 4.10 Last stage of a centrifugal compressor operating in stonewall. The high pressure gradient at the volute cutoff can excite all impeller natural frequencies and result in impeller failure. *(Sorokes JM, et al. The consequences of compressor operation in overload. In: Proceedings of the thirty-fifth turbomachinery symposium; 2006. Courtesy of the Dresser-Rand business, part of Siemens Oil & Gas.)*

FIG. 4.11 Last stage impeller failure after operating in stonewall for less than 40 h.

SURGE

Surge flow has been defined as peak head [7]. Below the surge point, head decreases with a decrease in flow (Fig. 4.2).

Surge is especially damaging to a compressor and must be avoided. During surge, flow reversal occurs resulting in reverse bending in nearly all compressor components. The higher the pressure or energy level, the more damaging the surge forces will be.

As flow is reduced at constant speed, the magnitude of V_{rel} decreases proportionally, causing the flow angle to decrease (see Figs. 4.5, 4.12, and 4.13). Additionally, the incidence angle is increased (Fig. 4.13).

The smaller the flow angle α, the longer the flow path of a given gas particle from the impeller tip to the diffuser outside diameter. When angle α becomes small enough, and the diffuser flow path long enough, the flow momentum of the gas is dissipated by the diffuser walls by friction to the point where the frictional forces are increasing (versus flow reduction) faster than the head is increasing (versus decreasing flow).

The high losses associated with low flow (see Fig. 4.13) are partly caused by a poor incidence angle i, which can result in flow separation at the low-pressure side of the blade leading edge. This flow separation frequently starts at one or more blades and continuously shifts around the impeller blades (or blade row in an axial compressor). This occurs at relatively low speeds just before full surge occurs. At higher speeds, the compressor generally goes directly from stable operation to flow separation on all blades and full reverse flow.

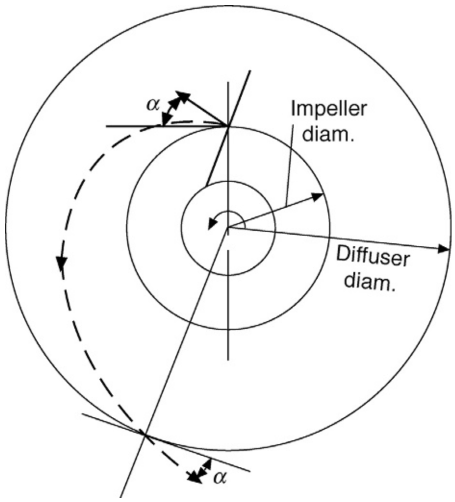


FIG. 4.12 Flow through a diffuser [17]. *(Used with permission of Elliott Company, Jeannette, PA.)*

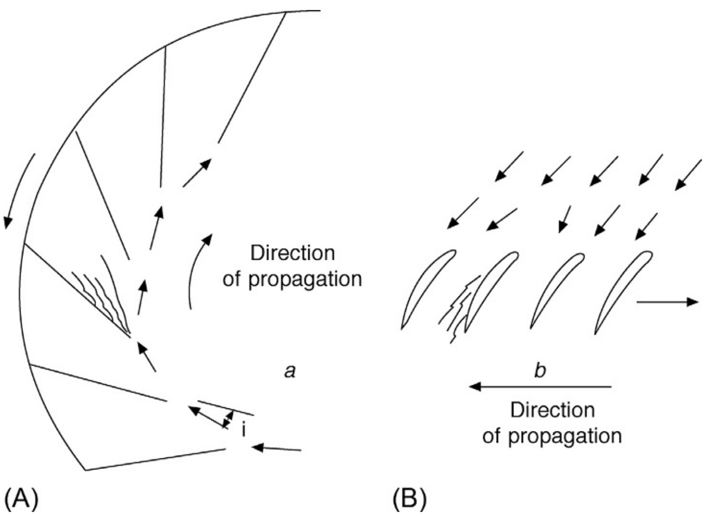


FIG. 4.13 Rotating stall: (A) Centrifugal compressor impeller. (B) Axial compressor blade pack.

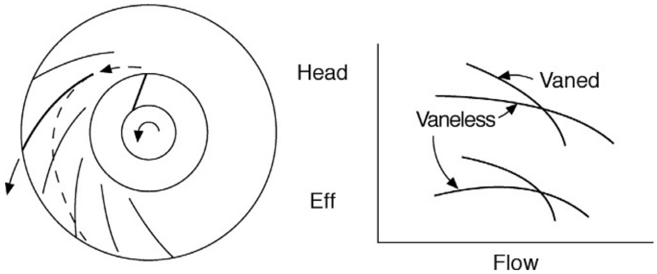


FIG. 4.14 Diffuser vanes. *(Data from Paluselli DA. Basic aerodynamics of centrifugal compressors. Jeannette, PA: Elliott Co. Hallock DC. Centrifugal compressors … the cause of the curve. Jeannette, PA: Elliott Co.; 1968.)*

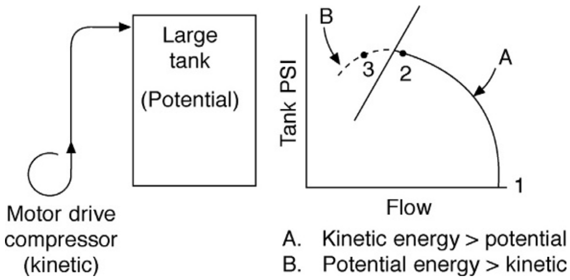


FIG. 4.15 Surge.

The flow separation plus the higher frictional losses result in a positively sloped curve. Since system resistance curves are also positively sloped, the system is unstable.

The point at which a compressor surges can be controlled somewhat by the designer adjusting the diffuser area to increase V_R and flow angle α. Of course, higher velocities result in higher frictional losses, so a designer must balance between desired surge point and stage efficiency during the design process.

The surge point is reduced by the addition of vanes in the diffuser (Fig. 4.14). The vanes shorten the flow path through the diffuser, reducing frictional losses and controlling the radial velocity component of the gas. Due to lower friction, head and efficiency are enhanced, but the operating range is reduced. Off-design operation rapidly changes the incidence angle to the vanes and flow separation occurs, resulting in the reduced operating range.

To better understand what is occurring during surge, visualize the simple system shown in Fig. 4.15. The system consists of a small motor-driven compressor delivering air to a relatively large tank. While in an idle state, the entire system is at ambient conditions. The instant the unit reaches design speed, the pressure in the tank is still zero (Point 1). As time passes, the pressure builds in the tank and flow is reduced due to increased resistance. Eventually Point 2 is reached where the pressure of the tank causes such a high backpressure on the compressor that flow through the impeller is significantly reduced. Much of the energy input is going to friction instead of building head. This is due to both the mismatch of inlet angle and the longer diffuser passage described earlier. Since this effect continues to build as flow is reduced, the slope of the head curve is reversed. As flow is reduced to Point 3, the head output of the compressor is also reduced. Since the pressure in the tank is still at Point 2, flow occurs from the tank to the compressor. Once the pressure in the tank is reduced (by reverse flow) to a level less than the head capability of the compressor, the process will then recover and the gas will flow from the compressor to the tank. This process will continue to repeat itself indefinitely.

OFF-DESIGN OPERATION[2]

Off-design operation of a compressor can dramatically affect the performance characteristic curve shape. Any change in inlet conditions can change the discharge pressure and gas horsepower as shown in Fig. 4.16 (see also Conceptualizing Head, Chapter 2).

In addition to changing the characteristic pressure and horsepower curves, the characteristic head curve also changes. This is due to volume ratio effects and equivalent speed effects.

2. Adapted from "Centrifugal compressors … the cause of the curve," D.C. Hallock, Elliott Company, Jeannette, PA, 1968, with permission [17].

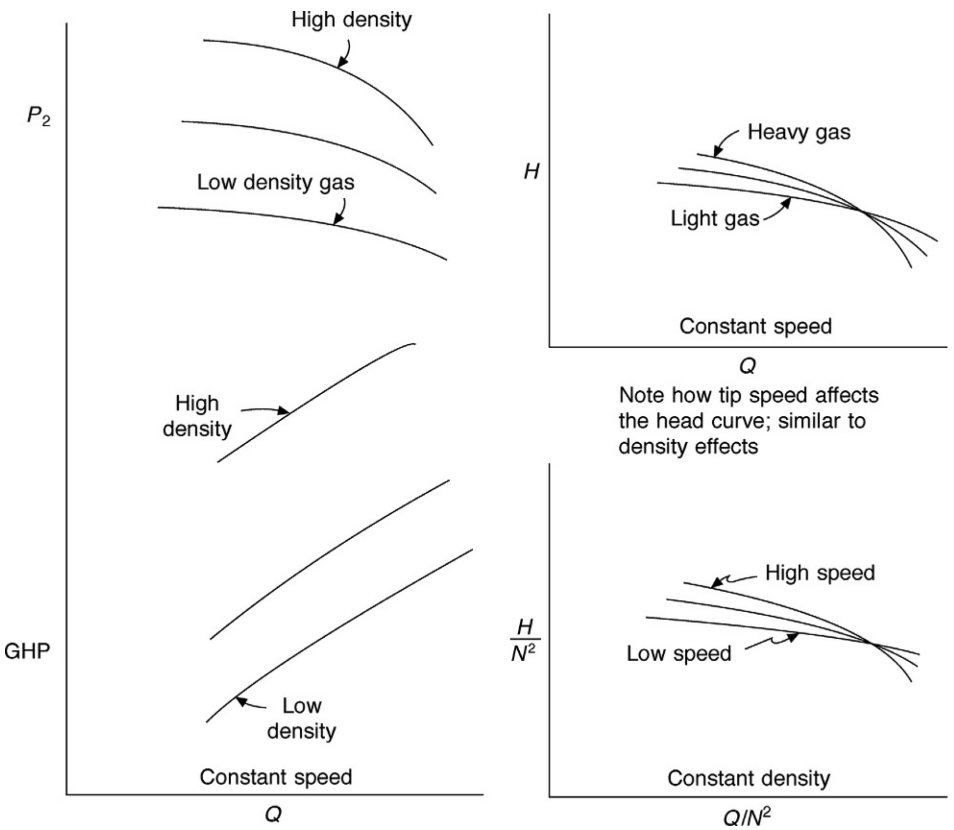


FIG. 4.16 Effect of varying inlet conditions at constant speed for a single-stage compressor. For a multistage compressor, the curve shape and operating range is further compounded by volume ratio effects. See Fig. 4.18. *(Data from Compressor refresher. Jeannette, PA: Elliott Co.; 1975. Hallock DC. Centrifugal compressors … the cause of the curve. Jeannette, PA: Elliott Co.; 1968.)*

If a constant discharge pressure is desired and gas conditions are changed (inlet pressure or temperature, mole weight change), a speed change is required. Since the curve shape changes with speed (higher losses at higher speeds), the head curve shape then changes (Fig. 4.17). This effect is further compounded by volume ratio effects (Fig. 4.18).

The head characteristics are a function of the acoustic velocity of the gas (see Fig. 4.9). Knowing this, it is most convenient to refer to some "constant" gas, and obtain an "equivalent tip speed." This reference constant is typically air at 80°F, since this is what most "developmental" testing uses as a test medium.

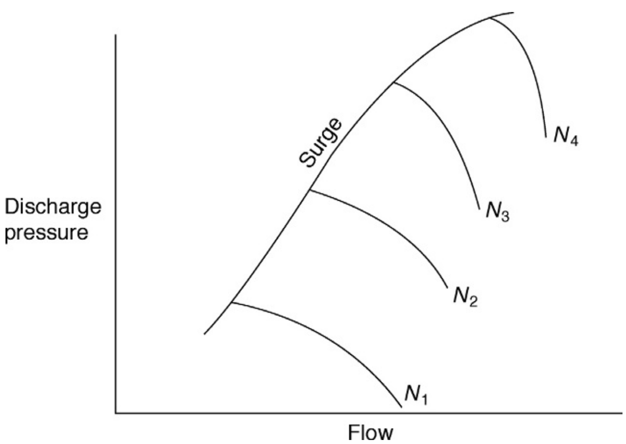


FIG. 4.17 Effect of speed change on compressor curve shape for a single-stage compressor.

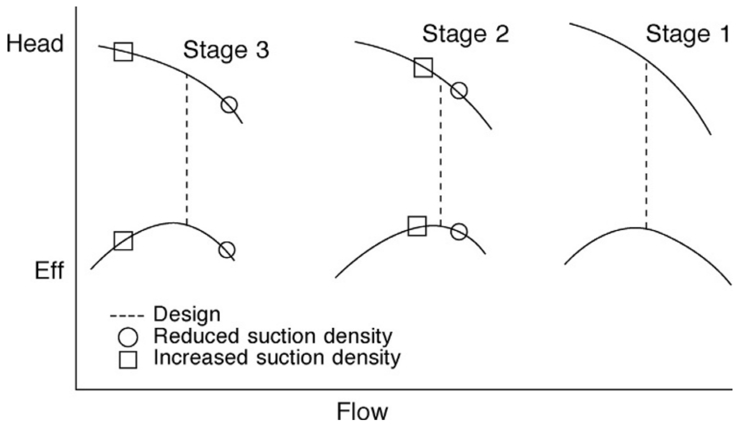


FIG. 4.18 Volume ratio effects. *(Data from Compressor refresher. Jeannette, PA: Elliott Co.; 1975.)*

As an example, we know the sonic velocity of air at 80°F is 1140 fps, and that of propylene at—40°F is 740 fps. If a compressor stage is operating at a mechanical tip speed of 780 fps on propylene at—40°F, the equivalent tip speed is

$$U_{eq} = 780 \times \frac{1140}{740}$$
$$= 1200\,\text{fps}$$

The stage characteristic head curve shape at 780 fps on propylene is therefore the same as 80°F air at 1200 fps. (Note: This is not exact. There is also an impeller tip volume ratio effect based on gas density that causes head and work input to change somewhat. An air and propylene test will not result in exactly the same head curve at constant equivalent speed, but for all practical purposes the results are close enough to be considered the same.).

In a multistage compressor, the "equivalent speed" effect is compounded by volume ratio effect (Figs. 4.18–4.20). If the gas density varies, the pressure rise and volume ratio will also vary. This will feed a different flow rate to the second stage. The effect on following stages will be compounded. The end result is a premature choke and surge (a shorter operating range).

ADJUSTABLE VANES

Adjustable vanes can be used to extend the useful operating range of any compressor (see Figs. 4.21–4.23). Adjustable vanes are typically used on the first several stages of axial compressors since they are very sensitive to the gas angle of attack and have a very short operating range. Adjustable vanes are also very popular for single-stage centrifugal units and are occasionally used on multistage centrifugal compressors.

The range of operations is extended by changing the angle of attack on the inlet side of the impeller blade (Figs. 4.1 and 4.8). For the high-flow region, the angle of attack can be enhanced to eliminate flow separation and effectively increase the "throat" area of the impeller. This will increase the capacity of the impeller. Also, head will increase due to "against" rotation swirl (see Fig. 4.9). By adjusting the vanes to provide swirl in the direction of the impeller rotation, V_{rel} is reduced, reducing head (Fig. 4.22). Since the incidence angle is improved, frictional losses are improved and peak efficiency as well as peak head shifts toward reduced flow.

AXIAL COMPRESSORS[3]

The operating principles of the axial compressor differ significantly from those of the centrifugal in that the compressor characteristics are dependent on the lift and drag coefficients of the cascade of airfoil blades (Fig. 4.24). The nominal axial velocity of the gas is constant throughout the axial, while in the centrifugal compressor the gas is being accelerated and decelerated. The reduced wetted perimeter that the gas "sees" and the short flow path through the axial contribute to the improved overall efficiency.

3. Based on "Fundamentals of fluid flow as applied to the design of axial flow compressors and fans," W.K. Bodger and R.C. Jensen Carrier Corporation, 1954, with permission [1]; and "Axial compressor design philosophy," P. Whiteman, Elliott Company, Jeannette, PA, with permission [19].

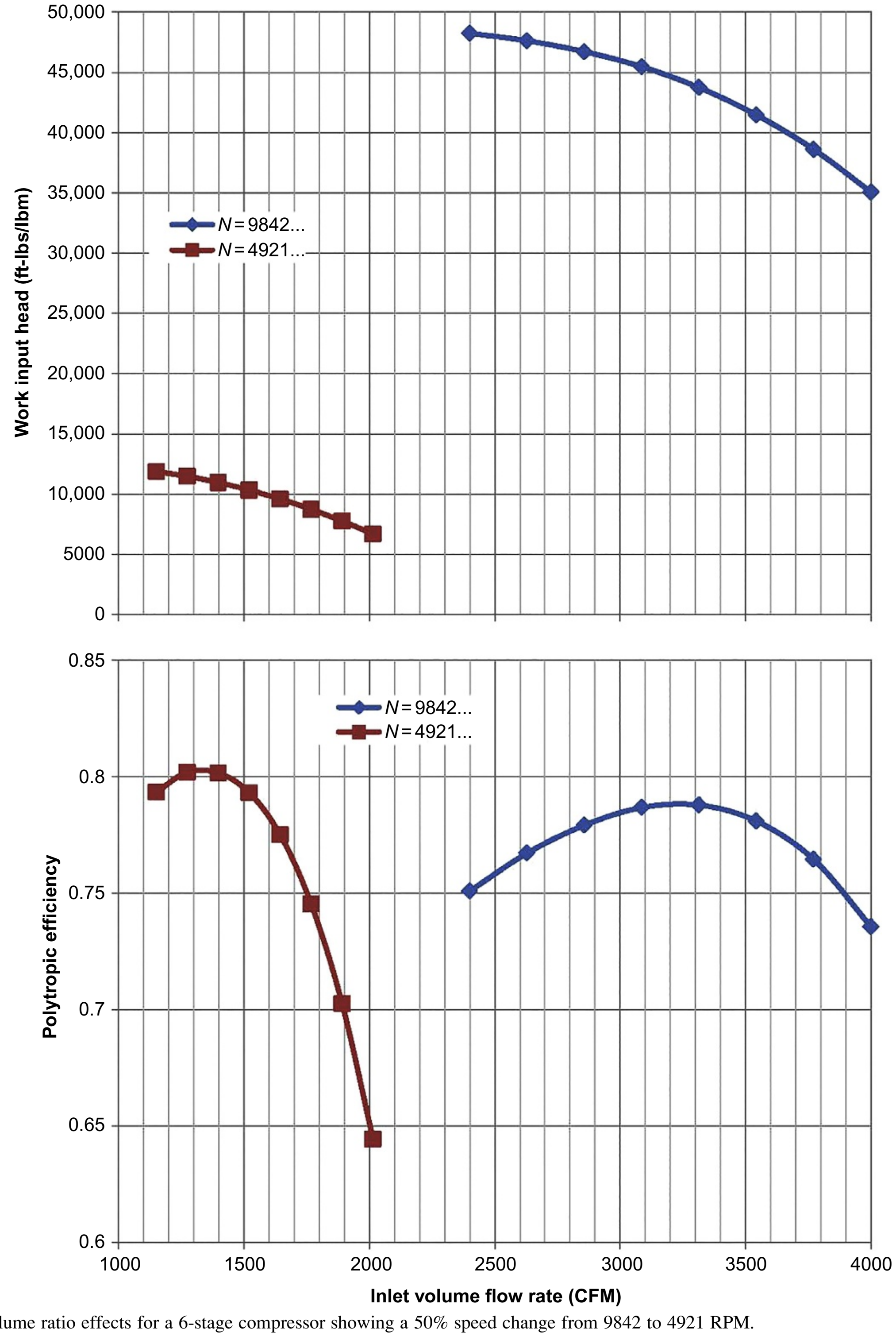


FIG. 4.19 Volume ratio effects for a 6-stage compressor showing a 50% speed change from 9842 to 4921 RPM.

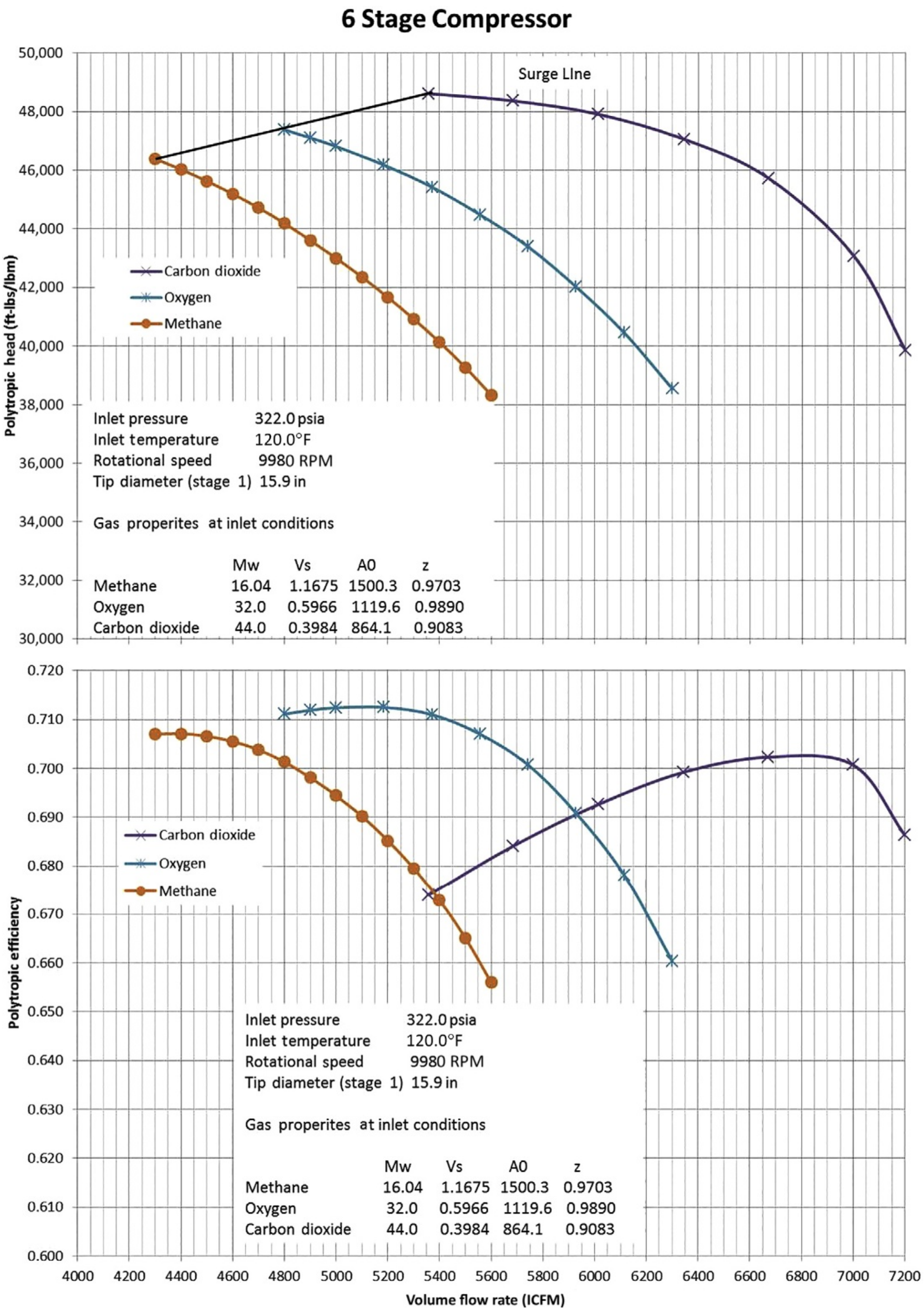


FIG. 4.20 Volume ratio effects for a 6-stage compressor showing performance curves for different gases at identical inlet conditions. Methane with a MW of 16, Oxygen with a MW of 32, and Carbon Dioxide with a MW of 44, all with inlet pressure at 322 psia, inlet temperature at 120°F, and speed of 9980 RPM.

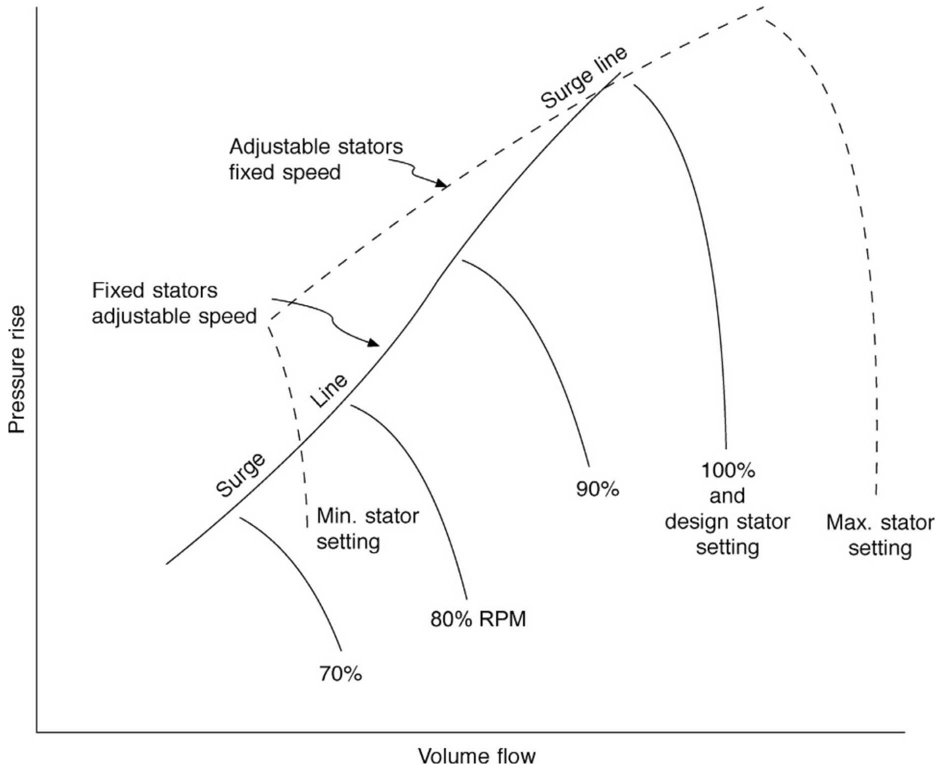


FIG. 4.21 Axial compressor performance map showing effect of adjustable stator vanes.

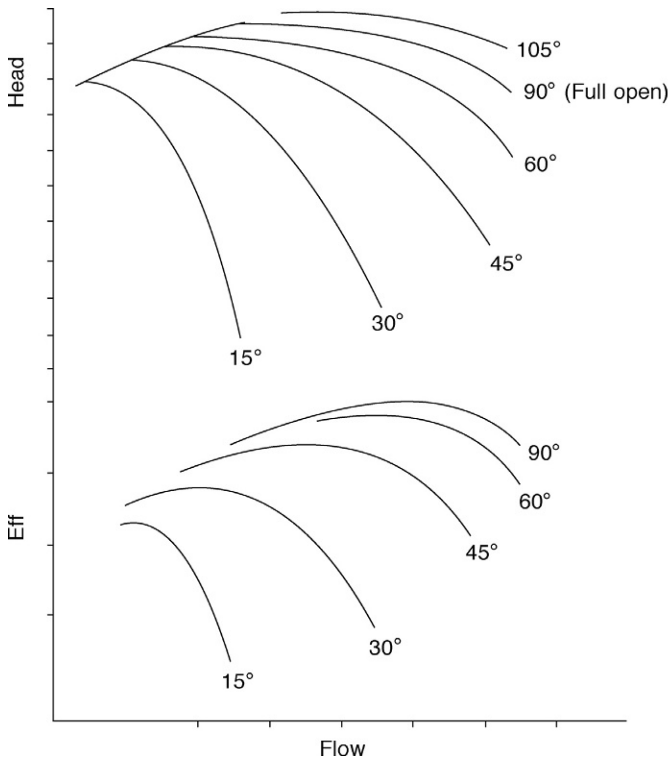


FIG. 4.22 Adjustable inlet guide vanes. Curve shows effect of various vane positions on head and efficiency for single-stage centrifugal compressor.

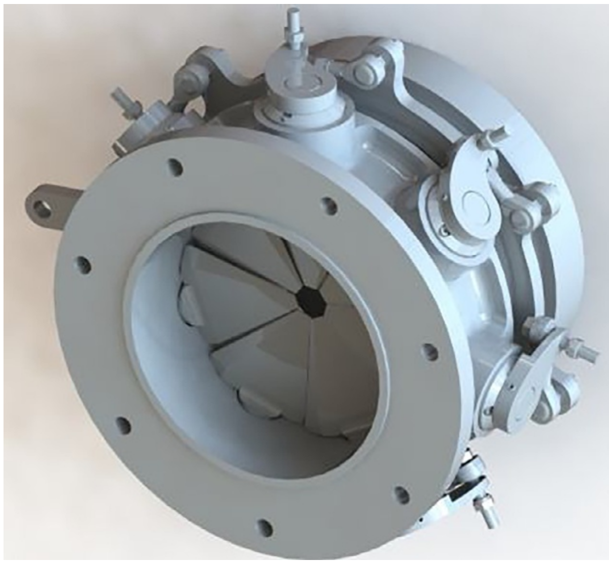


FIG. 4.23 Photo shows inlet guide vanes used on the first stage of a multistage centrifugal compressor. *(Photo courtesy of Sulzer-Escher Wyss Ltd.)*

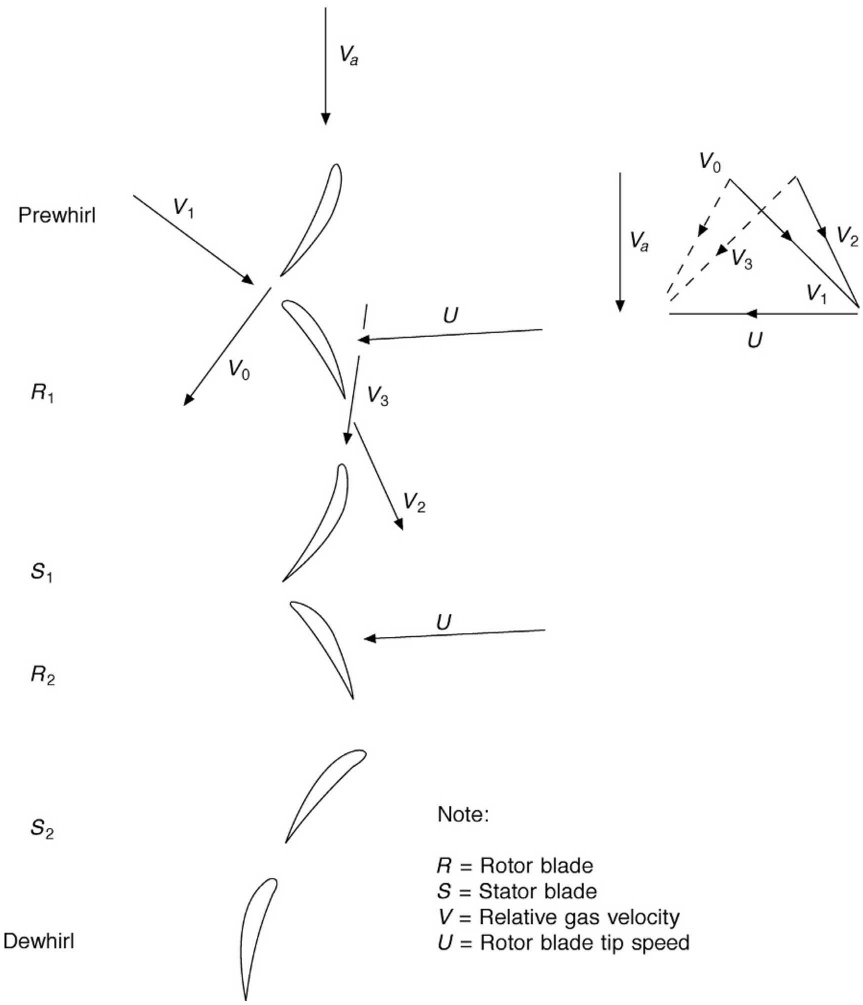


FIG. 4.24 Axial compressor vector diagram.

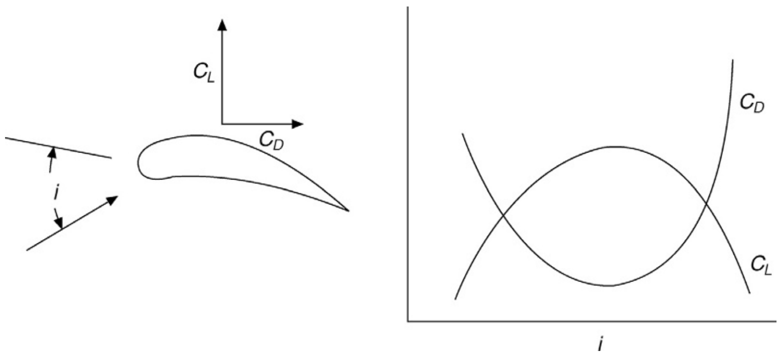


FIG. 4.25 Axial compressor staging is based on airfoil lift and drag coefficients. *(Data from Bodger WK, Jensen RC. Fundamentals of fluid flow as applied to the design of axial flow compressors and fans. Syracuse, NY/Jeannette, PA: Carrier Corp./Elliott Co.; 1954.)*

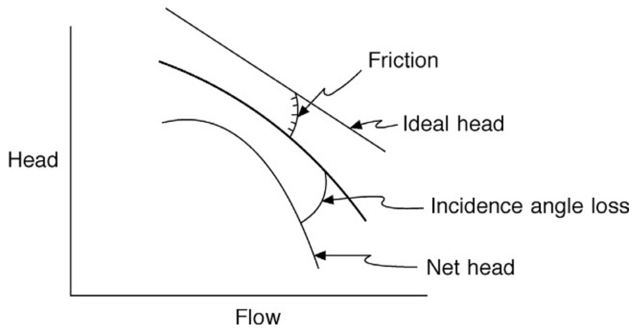


FIG. 4.26 Losses in an axial compressor stage are due to friction and incidence angle loss. *(Data from Bodger WK, Jensen RC. Fundamentals of fluid flow as applied to the design of axial flow compressors and fans. Syracuse, NY/Jeannette, PA: Carrier Corp./Elliott Co.; 1954.)*

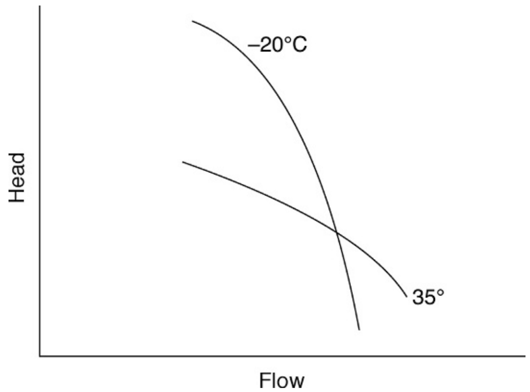


FIG. 4.27 Effect of suction temperature change on axial performance curve. *(Data from Whiteman P. Axial compressor design philosophy. Jeannette, PA: Elliott Co.)*

Fig. 4.25 shows general lift and drag characteristics of an airfoil. This is fine for an airplane wing, but for a compressor this information must be in the form of head and inlet flow as in Fig. 4.26.

Because relatively low-pressure rise is available from a single axial stage, relatively high speeds are utilized to maximize its effectiveness. Mach numbers are relatively high, so any change in inlet conditions, such as suction temperature, can dramatically affect the performance curve (Fig. 4.27).

Since Mach number, $M = V/a$, and sonic velocity, $a = \sqrt{kgRT}$, it is clear that for a given gas, Mach number is directly related to the temperature of the gas.

Reaction

Reaction is the degree of pressure rise in the rotating vanes versus overall pressure rise of the stage. Fifty percent reaction is the most popular, as this gives the best overall efficiency. In a 50% reaction stage, an equal amount of diffusion—or pressure rise—takes place across the rotating and stationary blades. In a 100% reaction stage, the total pressure rise is in the rotating blade while the stator simply acts as a turning vane (Fig. 4.28).

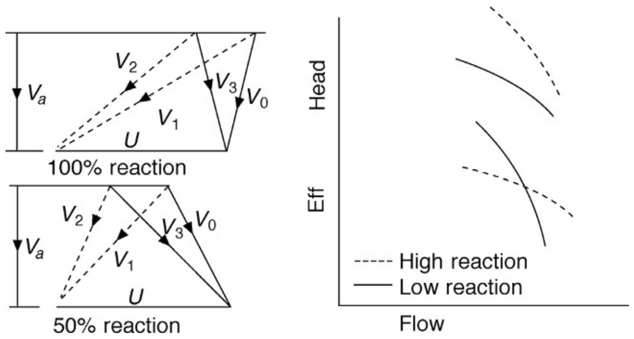


FIG. 4.28 Reaction. *(Data from Whiteman P. Axial compressor design philosophy. Jeannette, PA: Elliott Co.)*

Surge

From Fig. 4.25, it can be seen that a low angle of attack (low flow) results in high drag (low efficiency) and a low coefficient of lift (low head). As with the centrifugal compressor, the positive-sloped portion of the curve is unusable since this area of operation gives an unstable system and reverse flow occurs. Surge is especially damaging to an axial compressor because of the relatively large mass flow rate of gas and relatively thin blades. Besides the problem of reverse bending stresses and eventual fatigue, there is a problem of thermal growth. During surge, discharge gas is being forced back through the compressor and then recompressed. The compressor is now compressing heated gas and temperatures rise quickly, causing the blades to grow, eventually resulting in a rub.

At reduced speeds (70% of design or lower), rotating stall is almost unavoidable (Fig. 4.29). However, since rotating stall is only localized and temporary, the overall system is stable, and steady through-flow occurs keeping the blades cool. Since energy levels are lower, blade fatigue is not a serious concern. Even so, it is wise to minimize operation at this point.

Choke Flutter

At high-flow rates, the incidence angle becomes reversed and will eventually cause flow separation. This condition is called choke flutter and can be very damaging to the compressor blades (Figs. 4.29–4.31).

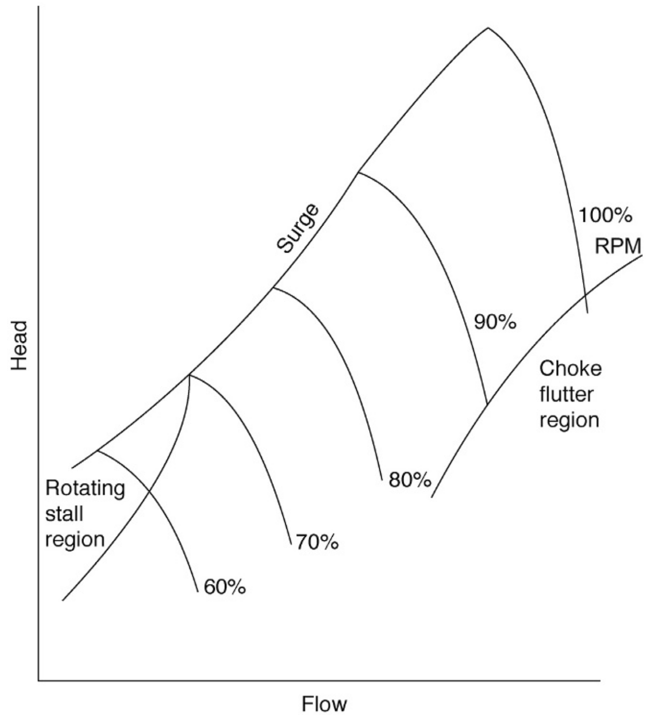


FIG. 4.29 Axial compressor performance map showing the choke flutter region. It is not easy to determine when choke flutter is occurring. Mounting vibration or strain gauge equipment on the stator vanes may be necessary to determine the presence of choke flutter.

FIG. 4.30 Axial compressor blade failure after operating in choke. An online performance monitoring system may have prevented this failure or at least have aided the troubleshooting efforts. *(Courtesy of Dakota Gasification, Beulah, North Dakota.)*

FIG. 4.31 Failed axial blade. *(Courtesy of Dakota Gasification, Beulah, North Dakota.)*

Part II

Application

This page intentionally left blank

Chapter 5

Equipment Selection

Whether purchasing a new piece of equipment or rerating an existing unit, it is important to know at least some basic design limits. You can rough out the preliminaries yourself before going to the equipment manufacturer.

When purchasing equipment, define more than a single operating point. Look at your process.

Know it well before specifying anything to the compressor manufacturer. Know the type of equipment available and what its characteristics are (Fig. 5.1; see also Fig. 1.7).A properly matched compressor is applied to the full range of expected operations. Sometimes, it may be wise to alter the process somewhat to provide a better match to the capabilities of the compressor.

NEW EQUIPMENT SELECTION

New equipment is the easiest to select since you are starting with a clean sheet of paper. It's just a matter of what equipment is available and how much can be spent. Don't forget that the control system and its logic can be more critical than the compressor selection. Remember, consider a "range" of operations, not a single point.

To start, however, select a single operating point. Using Eq. (2.33) and knowing the desired inlet and discharge conditions, determine the total head required for the compression equipment. To determine the number of compression stages required, some vendor data will be needed. Table 5.1 is an overview of what to expect. Note the higher head capability of the open impellers vs. the covered impellers. The cover increases the impeller stress for a given speed. Thus, the open impellers can operate at higher speeds and therefore provide higher head (Fig. 5.2). Since open impellers require close clearance to the upstream inlet piece or diaphragm (Fig. 11.6), use of open impellers is usually limited to single-stage compressors or to the first one or two stages of a multistage centrifugal compressor. Open impellers have also been used as the last stage in a multistage axial compressor.

All equipment that you select, regardless of the manufacturer, will have to be matched to existing available equipment, whether off the shelf or custom built. Even with custom-built equipment, the manufacturer generally designs a compressor using a building-block approach of existing components. It is rare, and expensive, to build a new piece of equipment from scratch for a given application.

Custom-design equipment is usually made up of several "families" of compression stages. There could be one family for high head, and another for high efficiency, and a middle-of-the-road family group of compression stages. These families are then scaled to various frame sizes to offer a wide selection, making it possible to customize for a given need by custom selection of standard components.

Although the families generally consist of 20–30 impeller sizes for a given frame size, this can be increased to an infinite number to enhance selectability. On one hand, it is important to keep this number to a minimum in order to minimize the number of drawings, castings, stampings, and performance records. Although this is a good idea, it is possible with the computerized equipment available for analysis, drafting, and manufacturing to have an infinite number of compression stages per compression family.

Although manufacturers' frame sizes are well-established standards, it is feasible that an infinite number of frame sizes could be made available with use of fabricated components, FEM analysis, and CAD. What this means is that a compression stage can be selected exactly at its peak efficiency instead of compromising and selecting the nearest frame size and the nearest impeller size (Fig. 5.3).

There is still the drawback that while the compressor may be very efficient at the design point, efficiency drops when operating at other than design conditions. To compensate for this, when purchasing a new compressor, specify a *range* of operations. Note the full operating range of flows, pressures, temperatures, and MW that you expect. Although some process conditions may call for continuous operation at one point, it is common to see some variation in operating conditions.

Compressor Performance. https://doi.org/10.1016/B978-0-12-814219-6.00005-2

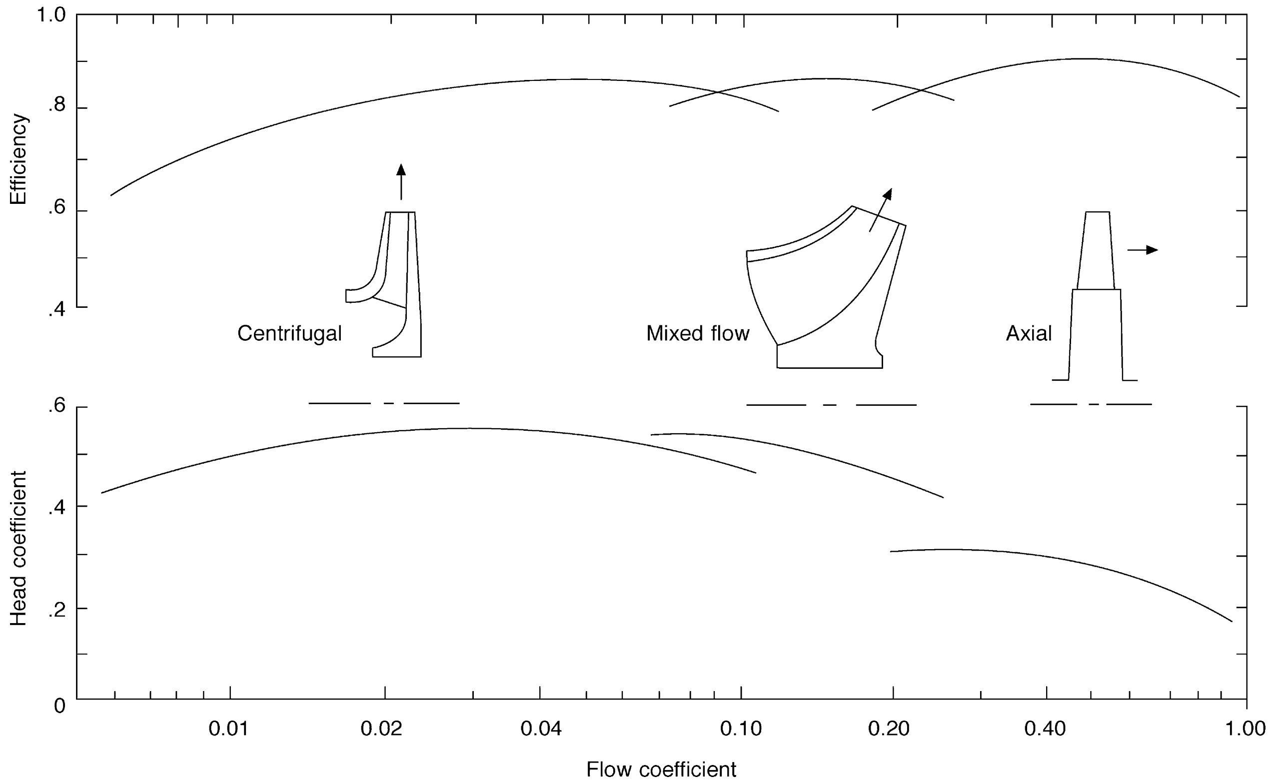


FIG. 5.1 Head and pressure capabilities for various compression elements.

TABLE 5.1 Approximate Per Stage Head Capability of Various Compression Elements

Stage	Head (ft-lb_f/lb_m)	r_p for Air
Centrifugal compressor		
Covered wheel	8000–12,000	1.3–1.5
Open wheel	30,000–60,000	2.6–8.9
Mixed flow		
Covered wheel	6000–8000	1.2–1.3
Open wheel	20,000–45,000	1.9–3.8
Axial compressor	4000–5000	1.15–1.20

Consider start-up and shut-down conditions. The entire operating range plus start-up and shut-down is important for mechanical (rotor dynamics, seals, auxiliary systems, etc.) as well as aerodynamic considerations. Include any purge gas that may be used on start-up (nitrogen, carbon dioxide, air, etc.).

Generally speaking, for a given flow, the smaller the frame size, the better. The smaller size means a lower price and usually a better efficiency. Look at Fig. 1.6. From this, it can be seen that there is an optimum flow for each compressor style. Fig. 5.1 shows this even more clearly. The smaller size compressor requires a higher flow coefficient. For even higher flow coefficient compression elements, a mixed-flow (Fig. 5.4) or an axial compressor is used.

For a single-stage compressor, it is wise to choose a midsized (flow coefficient, Eq. 2.60) stage to optimize efficiency. For multistage compressors, the first stage must be a high-flow stage so that use of very low-flow stages can be limited. Additionally, double-flow compressors can be used as the first section in a compressor, or as the first body in a string of equipment. Mechanical limitations must also be considered, such as maximum tip speed (stress limits), and stage spacing (critical speeds). High flow and high efficiency generally mean larger stage spacing (axial length), while high head and low flow generally mean reduced stage spacing. High flow and high efficiency design staging may require reduced speeds due

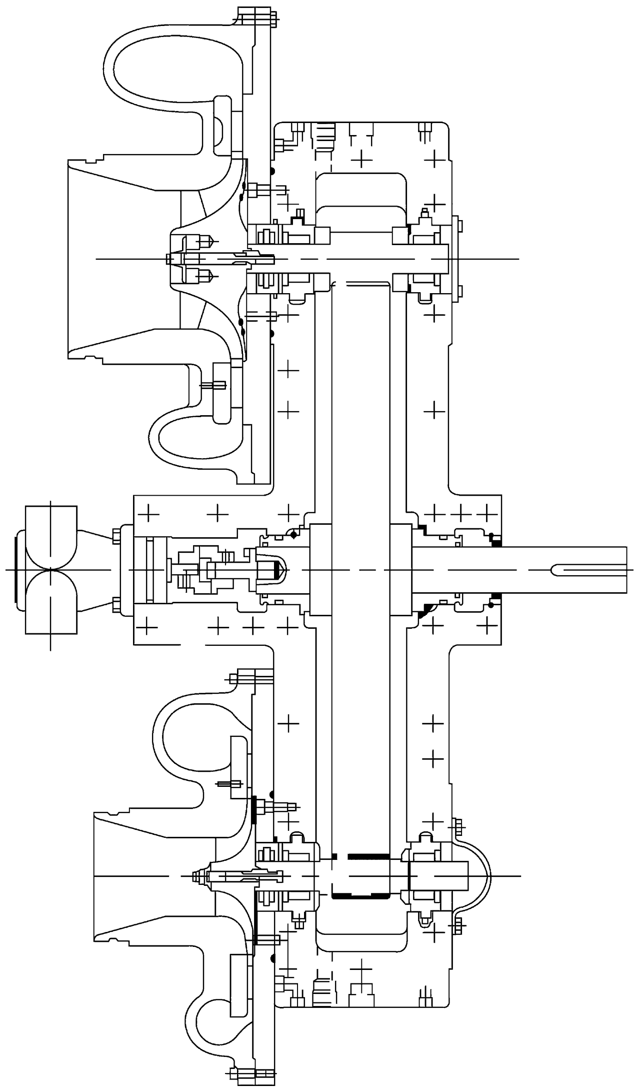


FIG. 5.2 High-speed, open-wheel centrifugal compressor. *(Used with permission of Elliott Company, Jeannette, PA.)*

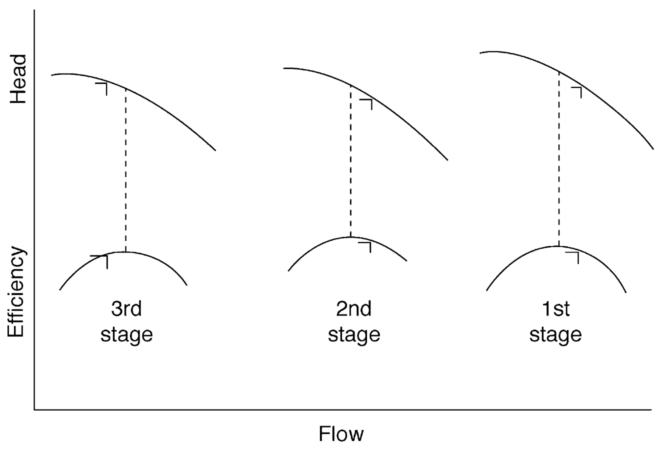


FIG. 5.3 Selection of a three-stage compressor utilizing a "fixed" family of compression stages. Note that in each case the design point is removed from the peak efficiency point. An infinite number of stages to select from would improve overall efficiency by allowing the design point to always be at maximum efficiency.

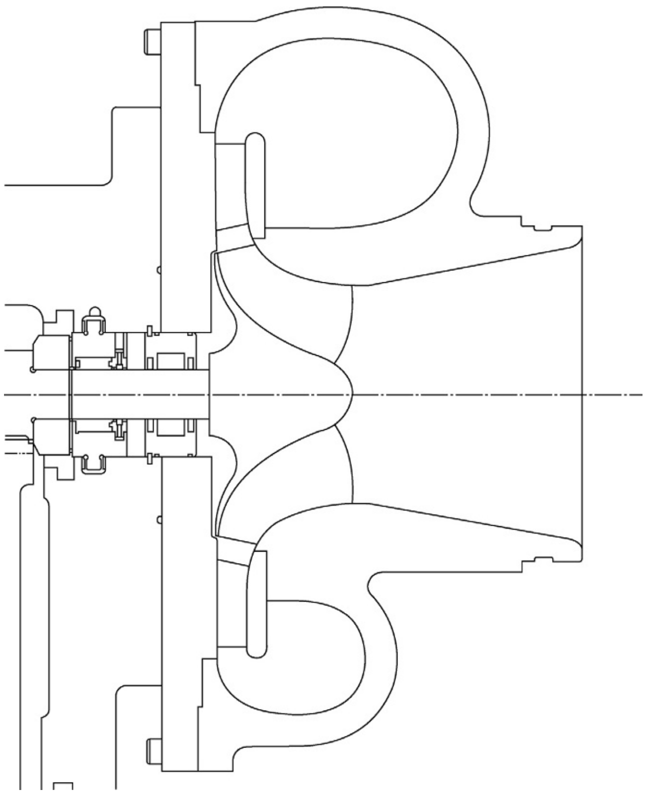


FIG. 5.4 Mixed-flow compressor. The midflow region (see also Fig. 5.1) is enhanced by the use of hybrid designs called mixed-flow stages, which use three-dimensional blade designs. In addition to improved efficiency, the mixed-flow compressor can result in a 30%–40% smaller package (volume and weight) for a given flow.

to higher stress levels encountered. Corrosive elements in the gas can also be a cause of limited speed. H_2S requires that controlled yield (80,000–90,000 psi) material be used for the impellers, thus limiting the operating speed.

For a given flow coefficient, larger equipment frame size means better efficiency. The larger size results in an improved "wetted perimeter" and thus reduced friction similar to using a larger pipe for fluid flow. It is better to use one large compressor rather than two small compressors in parallel. But the effect of stage size (flow coefficient, Fig. 5.1) may outweigh the frame-factor effect (Table 5.2). For example, assume an application where a compression stage for 60,000 CFM is required. Frame E in Table 5.2 would meet this need. However, for improved mechanical reliability, a better selection might be two D frames to operate in parallel. The single E frame would require less power if the same (scaled) impellers are used for both the D and E frame selections.

TABLE 5.2 Typical Centrifugal Compressor Frame Data

Frame	Nominal Flow Range (cfm)	Maximum Number Stages	Nominal Speed (RPM)	Nominal Polytropic Efficiency	Nominal H/N^2 Per Stage	Maximum Q/N
A	250–2500	8	15,000	0.75	3.7×10^{-5}	0.15
B	800–9000	8	11,500	0.78	7.5×10^{-5}	0.65
C	5000–25,000	8	8000	0.80	1.5×10^{-4}	2.9
D	15,000–35,000	8	6000	0.82	2.5×10^{-4}	5.5
E	30,000–70,000	8	5000	0.83	4.0×10^{-4}	12.0
F	55,000–125,000	8	3000	0.84	10×10^{-4}	50.0
G	100,000–170,000	8	2700	0.84	12×10^{-4}	65.0

SELECTION PROCEDURE

1. Start by determining the gas mixture properties. This can be done by referring to Tables 2.1 and A.1 (in Appendix A). Additionally, refer to Examples 7.1 and 7.4. If a gas properties program like Gas Flex is available, use it.
2. Next, calculate the actual inlet volume flow using Gas Flex for the conditions at the compressor suction flange. Normally expressed in cubic feet per minute, this flow is designated as ACFM (actual cubic feet per minute) or ICFM (inlet cubic feet per minute). Use the gas properties from Step 1 to make the conversion required.
3. Select the compressor frame size using Table 5.2.
4. Calculate total head required using Gas Flex. See Eq. (7.4).
5. Calculate the total number of stages required. Refer to the vendor literature on nominal head or H/N^2 values as shown in Table 5.2. Multiply this value by the nominal speed squared. Divide the total head by the per-stage head to approximate the number of stages.
6. Adjust the speed to obtain the discharge conditions desired by using fan laws, Eqs. (2.66), (2.67).
7. Gas horsepower should be corrected for balance piston leakage and mechanical losses, which should be available from vendor literature.

An outline of items for consideration during the selection process is provided in Table 5.3.

TABLE 5.3 Selection Outline

Rough out equipment yourself to get a "feel" for what you are buying
Specify a range of operations
If applicable, specify upratability for capacity and pressure
Consider shop and follow-up field performance testing
Provide for proper inlet piping
Consider abradable seals and diffuser coatings to maintain high efficiency
Review also the following:
Driver power
Foundation
Torque (shaft stress)
Lateral and torsional criticals
Couplings
Lube system
Casing pressure ratings
Impeller stress levels
Nozzle and volute velocities
At-speed balancing of the rotor
Provide for off-design conditions:
Air dryout
Nitrogen purge
Start-up
Shut-down
Peak load
Off load
Standby
Other

Example 5.1

N-Method

1. Given the following conditions,

$$\dot{M} = 1769 \text{ lb/min} \qquad MW = 29$$
$$P_1 = 80 \text{ psia} \qquad k = 1.4$$
$$T_1 = 90°\text{F } (550°\text{R}) \qquad Z = 1.0$$
$$P_2 = 225 \text{ psia}$$

2. Calculate inlet volume using Gas Flex.

$$v_1 = \frac{ZRT_1}{144P_1} = \frac{1.0(1545)(550)}{144(29)(80)} = 2.544\,\text{ft}^3/\text{lb} \tag{2.7}$$

$$Q = \dot{M} \times v_1 = 1769 \times 2.544 = 4500\,\text{ICFM}$$

3. Select compressor frame size.
   Based on an inlet volume of 4500 ICFM and knowing the required discharge pressure is 225 psia, select a B frame size from Table 5.2.
4. Calculate the required head using Gas Flex.
   Assume an efficiency of 0.76 from Table 5.2 and calculate the polytropic exponent.

$$\frac{n}{n-1} = \frac{k}{k-1}\eta_p = \frac{1.4}{0.4}(0.78) = 2.73$$

Calculate the overall head.

$$H = ZRT\left(\frac{n}{n-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right]$$
$$= 1.0\left(\frac{1545}{29}\right)(550)(2.73)\left[\left(\frac{225}{80}\right)^{0.3663} - 1\right]$$
$$= 36{,}837\,\frac{\text{ft-1b}_\text{f}}{\text{1b}_\text{m}}$$

Check the discharge temperature for a need to intercool (cool if $T_2 > 400°\text{F}$).

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(n-1)/n} = \left(\frac{225}{80}\right)^{0.3663} = 1.4605$$
$$T_2 = 550(1.475) = 803^\circ\text{R} = 343^\circ\text{F}$$

Isocooling is therefore not required.

5. Determine the number of casing stages.
   From Table 5.2, the nominal speed for a B frame is 11,500 RPM. Calculate the Q/N.

$$Q/N = \frac{4500}{11{,}500} = 0.391$$

From Table 5.2, $H/N^2 = 7.5 \times 10^{-5}$
H/stage would then be

$$H/N^2 \times N^2 = \left(7.5 \times 10^{-5}\right)(11{,}500)^2 = 9919\,\frac{\text{ft-1b}_\text{f}}{\text{1b}_\text{m}}$$

Determine approximate number of casing stages.

$$\text{Number of stages} = \frac{36{,}873}{9919} = 3.71$$

Four stages are required.

6. Adjust speed.
   Adjust the nominal speed according to the casing stages. Four stages must develop 37029 ft-$\text{lb}_\text{f}/\text{lb}_\text{m}$ or an average of 36,837/4 = 9209 ft-$\text{lb}_\text{f}/\text{lb}_\text{m}$ per stage.
   Using Fan Law relationships, adjust the speed.

$$H \propto N^2$$

$$N = N_{\text{NOM}}\left(\frac{H_{\text{REQ'D}}}{H}\right)^{1/2} = 11,500\left(\frac{9209}{9919}\right)^{1/2}$$

$$N = 11,081\,\text{RPM}$$

7. Calculate the approximate power.

$$\text{GHP} = \frac{\dot{M} \times H}{33,000 \times \eta_p} = \frac{1769 \times 36837}{33,000 \times 0.78} = 2532\,\text{HP}$$

Adjust for balance piston leakage.
2532 × 1.02 = 2583 HP
Add losses for bearings and seals (data from vendor literature).
SHP = 2583 + 65 = 2648 HP

RERATES[1]

Frequently, process requirements change due to market fluctuations, new concepts in processing, a depleting gas field, or other reasons. Because of this, the compressor can be called upon to operate in a range for which it was not originally designed.

One way to satisfy the needs of the new conditions is of course to buy new equipment. A rerate, however, can often provide the means of meeting these new conditions at significantly less cost than new equipment, since much of the hardware can be reused. Foundations and piping remain the same for a rerate thus saving time and money.

A rerate can be difficult in the sense that the Application Engineer is very limited. The casing size and bearing span is fixed. Capacity and pressure rise is therefore limited.

To determine the rerate feasibility of a compressor rerate, the following must be considered:

1. *Capacity*: Are the nozzles large enough to accept the new flow rate?
2. *Horsepower*: Can the motor, turbine, or gear handle the new horsepower?
3. *Pressure*: Can the compressor handle the new pressure rise? Is the casing able to meet the new pressure levels?
4. *Speed*: Can the rotor handle the new speed? What are the effects on rotor dynamics and impeller stress?

Capacity

The primary factor in considering the capacity limit is whether the nozzles will pass the required flow at a reasonable pressure drop. A good rule of thumb for inlet nozzles for air or light gas is 140 fps. Otherwise, use a velocity such that

$$\frac{P_v}{P_1} = 0.01$$

(see Fig. 6.17)

where

$$P_v = \frac{P_1 V_1^2}{2gZ_1RT_1}$$

This should be calculated using the area schedule at the inlet flange. For discharge nozzles, velocities can be about 35% higher. The discharge volute can also be the limiting factor. However, only the OEM can properly determine this.

---

1. Adapted from "Can you rerate your centrifugal compressor?," Ron Lapina, Elliott Company, Jeannette, PA, 1975, with permission [20].

Horsepower

The rebuilt compressor will require a power increase approximately proportional to the increase in weight flow (see Eq. 7.12). This means that an increase in weight flow of 20% will require an increase in horsepower of at least 20%.

Another consideration is pressure rise or head. Horsepower is directly proportional to head.

$$\text{GHP} = \frac{H_p \dot{M}}{\eta_p 33{,}000} \tag{7.12}$$

If an increase of 20% in weight flow is coupled with an increase in polytropic head of 20%, the power will rise by 44%.

$1.20 \times 1.20 = 1.44$

An additional 10% may be added to this figure for operation in the overload region.

Motors are generally not designed with very much extra capacity and therefore are not generally rebuilt for such power changes. A new motor may be required. Check the foundation. Turbines and gears can, like the compressor, be rerated and in some cases have been overdesigned and will be able to handle the power increase. Whatever your case may be, the driver power capability is a crucial point in the rerate analysis.

Pressure

The compressor, when originally manufactured, was hydrotested at 1.5 times the maximum operating or settleout pressure. If this pressure is exceeded, the compressor casing should be rehydrotested. Note that on multisectional compressors, each section may have been tested at different pressure levels. In such a case, each section must be reviewed separately. A rehydrotest should be completed if only one section exceeds the original design pressure.

The ability of the compressor to develop the required pressure rise must be investigated. This is done by calculating the required head using Eq. (7.4a). If the gas composition has changed, new values for Z, R, and n must be established (see sample problems in Chapter 7).

$$H_p = Z_1 R T_1 \left(\frac{n}{n-1}\right) \left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1 \right] \tag{7.4a}$$

For simplicity, assume projected efficiency to be similar to existing efficiency.

Use Eq. (2.67) to determine the new speed.

$$N_{\text{rerate}} = N_{\text{original}} \sqrt{H_{\text{rerate}}/H_{\text{original}}} \tag{5.1}$$

This speed may turn out to be too high for impeller stress or rotor dynamics. However, sometimes, the manufacturer can increase the head by means of selecting an impeller from a different family designed for high head. Also, these standard impeller designs can sometimes be modified by increasing the diameters. This concept can provide both improved efficiency and increased head as long as the operating speed does not cause excessive impeller stress.

If there is extra space in the compressor for an additional impeller, the head capability can be approximated by

$$H_{p\ rerate} = H_p \text{original}[(a+b)/a] \tag{5.2}$$

where

a = number of impellers in the original rotor

b = number of "blank stages" in the original rotor

Once you have applied this "blank stage factor," go back to the fan law, Eq. (2.67), to estimate the new speed required to meet rerate H_p.

Speed

Speed is a serious consideration. The new speed must not cause high stress in the impellers. The critical speeds of the rotor must be avoided and the rotor must be stable at the maximum operating speed.

It is not really possible for the field engineer to determine the safe operating speed for impellers. Only the manufacturer can determine this. There may be available a new impeller design or improved materials to handle the required speed

increase. However, as a starting point, you might use 1000–1100 fps maximum tip speed for covered wheels and 1200–1400 fps for open impellers.

Since the primary factors affecting the critical speeds are bearing span and shaft diameter, the critical speeds are somewhat fixed. However, new bearings or other factors may provide the extra margin to a critical speed if necessary. Again the manufacturer must determine this, along with rotor stability.

Also, remember that the shaft-end stress and torsional criticals must be reviewed. This is usually something left up to the OEM.

As most rerates are field retrofitted, the only testing is done at the site. Thus, there is always some question about how the unit will run. An at-speed balance is a good way to gain some added "insurance" that the compressor will operate successfully.

Example 5.2

Consider a "straight through" centrifugal compressor on a dry-air process, with the following data:

Inlet capacity = 11,000 icfm
Inlet temperature = 90°F
Rated inlet pressure = 14.5 psia
Rated discharge pressure = 55 psia
Rated power input = 1700 HP
Rated speed = 8100 RPM
Max. continuous speed = 8500 RPM
First critical speed = 4,800 RPM
Second critical speed = 10,800 RPM
Rated molecular weight = 28.97
$k = c_p/c_v = 1.4$
Max. discharge pressure = 65 psia

The largest wheel diameter is 22 in., and the cross-sectional drawing of the compressor indicates that the inlet nozzle diameter is 20 in.

The desired rerate is an increase of the inlet capacity to 12,300 icfm and an increase of the discharge pressure to 60 psia, with all other inlet conditions unchanged.

1. Compute the inlet velocity, based on the new inlet volume flow:

$$V_a = 3.06\left(\frac{Q}{D^2}\right) = 3.06\left(\frac{12{,}300}{20^2}\right) = 94\,\text{ft/s}$$

Since this is an acceptable inlet velocity (Fig. 6.17), the proposed capacity is feasible.

2. Since the rated inlet conditions have not changed, the increase in weight flow will be proportional to the increase in volume flow; therefore, the power requirement due to the change in volume flow will increase by the same proportion:

$$\begin{aligned}\frac{\text{GHP}_{\text{rerate}}}{\text{GHP}_{\text{original}}} &= \frac{\dot{M}_{\text{rerate}}}{\dot{M}_{\text{original}}} = \frac{Q_{\text{rerate}}}{Q_{\text{original}}}\\ &= \frac{12{,}300}{11{,}000} = 1.12\\ \text{GHP}_{\text{rerate}} &= 1.12\,\text{GHP}_{\text{original}}\\ &= 1.12(1700) = 1900\,\text{HP}\end{aligned}$$

Note that, up to this point, the driver will have to be capable of at least
(1.1)(1900) = 2100 HP
plus 2% excess horsepower if a gear is involved.

3. Since the nameplate maximum-discharge-pressure is 65 psia, the compressor will not have to be hydrostatically retested, provided that the process will not allow the value of 65 psia to be exceeded.
4. The approximate polytropic head can now be calculated for both the original and the rerate conditions from Eq. (7.4).

Original:

$$n/(n-1) = k/(k-1)\eta_p = (1.4/1.4-1)(0.76) = 2.66$$
$$H_p = Z_1 RT_1 (n/(n-1))\left[(P_2/P_1)^{(n-1)/n} - 1\right]$$
$$= (1.0)(1,545/28.97)(550)(2.66)\left[(55/14.5)^{1/2.66} - 1\right]$$
$$= 50,700$$

Rerate:

$$H_p = (1.0)(1545/28.97)(550)(2.66)\left[(60/14.5)^{1/2.66} - 1\right]$$
$$= 55,000$$

The new required speed can be determined from the fan law, Eq. (2.67).

$$N_{\text{rerate}} = N_{\text{original}}\sqrt{H_{p\,\text{rerate}}/H_{p\,\text{original}}}$$
$$= 8100\sqrt{55,000/50,700} = 8,440\,\text{RPM} \tag{5.3}$$

The largest wheel diameter is 22 in; therefore, from Eq. (5.6)

$$U = \frac{\pi dN}{720} = \frac{\pi(22)(8440)}{720} = 810\,\text{ft/s}$$

The new rotational speed results in a satisfactory mechanical tip speed. API states that the second critical speed must be 20% above the highest operating speed. Assuming that the new rerate speed is the highest for the new process, the second critical speed must be at least (1.2)(8440) = 10,130 RPM.

The second critical speed (10,800 RPM) is higher than that required; therefore, the rotational speed is feasible.

Be sure to check the method used for determining the critical speed. The first critical speed should have been verified in operation if a flexible rotor is used. The second or higher critical(s) may not be accurately known, and the only way to determine this accurately is with a rotor response analysis (a critical speed analysis that considers bearing and support stiffness). While this is being done, rotor stability should also be reviewed.

5. The total increase in gas horsepower can now be determined. The new horsepower will be proportional to the increase in polytropic head and weight flow (in this case, volume flow).

$$\frac{\text{GHP}_{\text{rerate}}}{\text{GHP}_{\text{original}}} = \left(\frac{H_{p\,\text{rerate}}}{H_{p\,\text{original}}}\right)\left(\frac{Q_{\text{rerate}}}{Q_{\text{original}}}\right)$$
$$= \left(\frac{55,000}{50,700}\right)\left(\frac{12,300}{11,000}\right) = 1.21 \tag{5.4}$$
$$\text{GHP}_{\text{rerate}} = (1.21)(1700) = 2060\,\text{HP}$$

The driver must therefore be capable of
(1.1)(2060) = 2270 HP
plus 2% if a gear is involved.

Since the inlet velocity, maximum operating pressure and required rotational speed are within satisfactory limits, the rerate is feasible.

SHOP TEST[2]

The best way to assure equipment quality is to have the compressor performance tested prior to shipment from the factory (Fig. 5.5). Although the performance test can be done in the field once the compressor is installed, the quality of a field test is generally less than a shop test and it is difficult to make corrections if necessary, as timing is always tight during the installation phase.

This is true for all compressors, even "off-the-shelf" varieties. Although the design may be proven, parts can be mismachined or improperly assembled. Custom-built compressors have the added risk of errors in application or design engineering.

ASME Power Test Code (PTC10-1997) has defined two types of performance tests:

Type 1. This is a test run on the design gas near design conditions. Generally, this applies to air compressors. This also applies to full-load shop tests or field testing.

2. Adapted from "Compressor refresher," Elliott Company, Jeannette, PA, 1975, with permission [11].

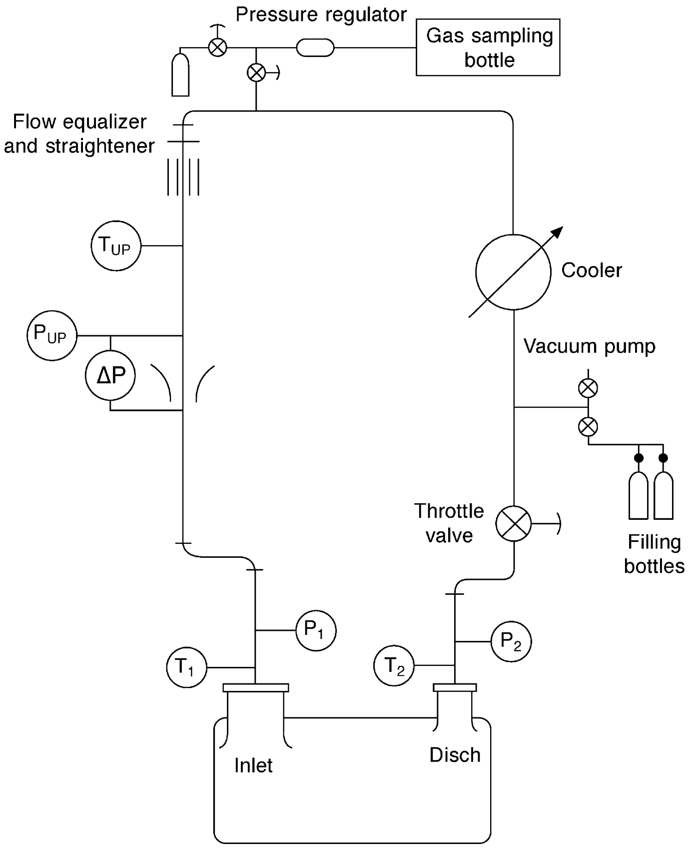


FIG. 5.5 Typical shop test [11]. *(Used with permission of Elliott Company, Jeannette, PA.)*

Type 2. When using the design gas for testing is not practical and a substitute gas is used for the test, a type two test is conducted. The gas used for the type 2 test does not have to follow the perfect gas laws.

The type 1 test is relatively simple to complete. Be certain of obtaining good data and complete calculations per PTC 10 (refer to Chapter 7, "Field Performance Testing"). A type 2 test requires some special considerations upfront. In determining the proper equivalent conditions, the following three items must be reviewed: (1) volume ratio, (2) Mach number, and (3) Reynolds number.

Unfortunately, it is not always possible to match all three parameters with an equivalent test, so some compromise must be made.

On a multistage compressor, volume ratio is of utmost concern (see Fig. 4.18), and should be closely matched. Per API, the variation between test and actual should not exceed 5%. At other than design volume ratio, downstream stages will not see the design flow and the overall curve will be affected.

To assure the test volume ratio equals the design volume ratio, use Eq. (5.5).

$$N_t = N_d \sqrt{\frac{Z_t \mathrm{MW}_d T_t}{Z_d \mathrm{MW}_t T_d}} \sqrt{\frac{[n/(n-1)]_t \left[(P_2/P_1)^{(n-1)/n} - 1\right]_t}{[n/(n-1)]_d [(P_2/P_1)]^{(n-1)/n} - 1]_d}} \tag{5.5}$$

where N is the speed, d is the design conditions, and t is the test conditions.

Note that N_t is proportional to $\mathrm{MW}_d/\mathrm{MW}_t$. Assuming a design MW of 70 and test MW of 28 (other values fixed), the test speed would be much higher than design. This normally puts the speed beyond the mechanical limits, demonstrating why air or nitrogen cannot be used for equivalent tests for a machine designed for compressing a heavy gas like chlorine or propane.

As demonstrated in Fig. 4.9, Mach number determines the capacity of a compressor. Therefore, it is critical to assure the Mach number is within 5% of the design Mach number. The equation for determining test speed to duplicate Mach number is

$$N_t = N_d \sqrt{\frac{(\mathrm{kg}ZRT)_t}{(\mathrm{kg}ZRT)_d}} \tag{5.6}$$

It is suggested that the test speed slightly exceeds the Mach number test speed for conservative results [11]. If possible, select a test gas with a k value near the design k value.

With the test speed set by volume ratio and Mach number, we are left with a variation in Reynolds number. According to ASME PTC 10-1997, the performance of a compressor is affected by the Machine Reynolds number. Frictional losses in the internal flow passages vary in a manner similar to friction losses in pipes or other flow channels. If the Machine Reynolds number at test operating conditions differs from that at specified operating conditions, a correction to the test efficiency is necessary to properly predict the performance of the compressor. The ASME Power Test code PTC 10-1997 provides correction factors for variations in Reynolds number.

For a Type 1 test, the data are used directly to determine field performance. For a Type 2 test, the following is needed to convert test data to actual field conditions.

Capacity

$$Q_d = Q_t(N_d/N_t) \tag{5.7}$$

where N is the speed and Q is the compressor suction flow.

Efficiency

Since frictional losses in the compressor are a function of the Machine Reynolds number, it is appropriate to apply the correction to the quantity $(1-\eta)$. The magnitude of the correction is a function of both the Machine Reynolds number ratio and the absolute value of the Machine Reynolds number, with increasing effect as the Machine Reynolds number decreases.

The correction to be applied is as follows:

(a). *For centrifugal compressors*

$$\left(1-\eta_p\right)_d = \left(1-\eta_p\right)_t \left(\frac{RA_d}{RA_t}\right)\left(\frac{RB_d}{RB_t}\right) \tag{5.8}$$

$$RA = 0.066 + 0.934\left[\left(\frac{4.8\times 10^6 \times b}{\mathrm{Rem}}\right]^{RC} \tag{5.9}$$

$$RB = \frac{\log\left(0.000125 + \dfrac{13.67}{\mathrm{Rem}}\right)}{\log\left(\in + \dfrac{13.67}{\mathrm{Rem}}\right)} \tag{5.10}$$

$$RC = \frac{0.988}{\mathrm{Rem}^{0.243}} \tag{5.11}$$

where

Rem = Machine Reynolds number

= Ub/v'

U = tip speed of first stage impeller or first stage axial blade, ft/s.

b = blade height at the tip of the first stage centrifugal impeller, or the cord at the tip of the first stage axial rotor blade, ft.

v' = kinematic viscosity of the gas at compressor inlet conditions, ft^2/s.

ϵ = average surface roughness of the flow passage, in.

(b). *For axial compressors*

The correction for axial compressors is a function of the Machine Reynolds number ratio.

$$\left(1-\eta_p\right)_d=\left(1-\eta_p\right)_t\left(\frac{\mathrm{Rem}_t}{\mathrm{Rem}_d}\right)^{(0.2)} \tag{5.12}$$

The limitations of Appendix D apply.

Head

The polytropic head coefficient is corrected for Machine Reynolds number in the same ratio as the efficiency.

$$\mathrm{Rem}_{\mathrm{corr}}=\frac{\mu_{p_d}}{\mu_{P_t}}=\frac{\eta_{P_d}}{\eta_{P_t}} \tag{5.13}$$

or, for polytropic head

$$H_{Pd}=H_{Pt}\left(\frac{\eta_d}{\eta_t}\right)\left(\frac{N_d}{N_t}\right)^2 \tag{5.14}$$

This page intentionally left blank

Chapter 6

Operation

Although some compressor installations may be relatively simple and knowing the location of the start and stop button is sufficient for operation, it is always best to understand both the operating characteristics (Fig. 6.1) of the compressor and how it interacts with the process. Knowing how to properly install, operate, and control a compressor can add years to the useful operating life of the compressor as well as minimize operating costs.

PERFORMANCE CURVES

Before operating a compressor, it is important to become familiar with the performance curves.

Head/Efficiency

The most versatile of all is the head and efficiency versus flow curve. This is the basic curve required for describing any compressor (Fig. 6.2). Although this curve is useful for small changes in inlet gas conditions, its flexibility is limited. See Fig. 4.14.

Horsepower/Discharge Pressure

This curve is the most directly usable curve. As long as inlet conditions are matched to the curve, the curve can be read directly without any calculations. The drawback, however, is that inlet conditions are frequently different from the curve, so quite often this curve is useless (Fig. 6.3).

Pressure Ratio/Efficiency

Pressure ratio and efficiency plotted versus inlet flow is probably the most useful curve for quickly "ballparking" compressor performance, since inlet pressure is a variable. Note that this curve is only good for the specified suction temperature (Fig. 6.4).

If a head curve has been plotted, it is easy to develop a pressure ratio curve using Eq. (7.4b).

Nomograph Plots

Many years ago, before PCs were in common use by engineers, nomographs were a favorite tool of the engineer to save time-consuming tedious calculations. They have been useful even with performance calculations (Fig. 6.5). However, the accuracy of these nomographs could be very limited. Nomographs were good for rough, quick estimates on the effect of inlet condition changes, but not much more. With all the software available today like CompAero and others, engineers go through the detailed calculations necessary to obtain very accurate head, work & efficiency curves for off-design conditions.

Whichever curve is used, it is important to realize that the curves do vary with gas conditions as described earlier. Although the head curve is reasonably accurate for single-stage compressors, significant error can occur for multistage compressors when inlet gas conditions are varied from that stated on the performance curve (see "Off-design Operation," Chapter 4).

When plotting performance data for a variable speed compressor, it is wise to make a plot of H/N^2 (or r_p/N^2) versus Q/N. This will minimize the effects of small (2%–3%) speed changes (see Fig. 6.6).

Compressor Performance. https://doi.org/10.1016/B978-0-12-814219-6.00006-4

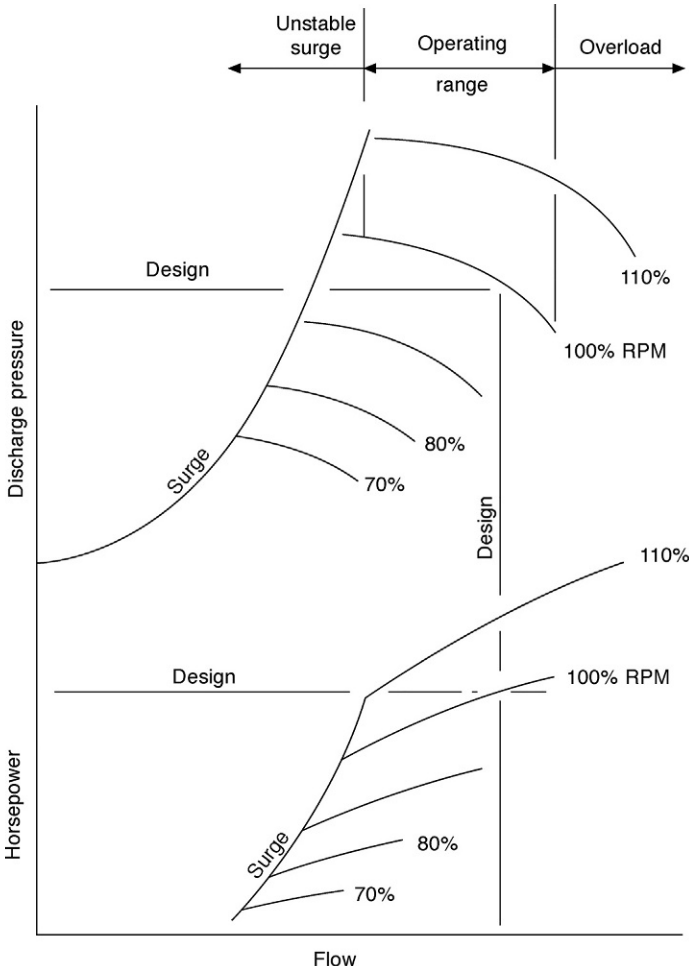


FIG. 6.1 Typical performance map.

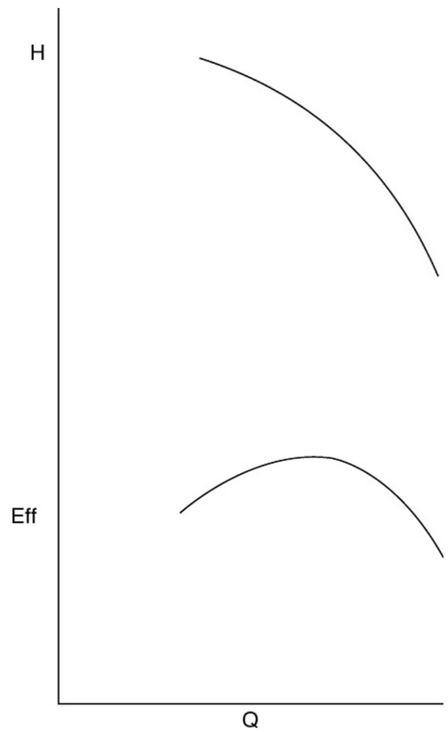


FIG. 6.2 Head/efficiency performance curve.

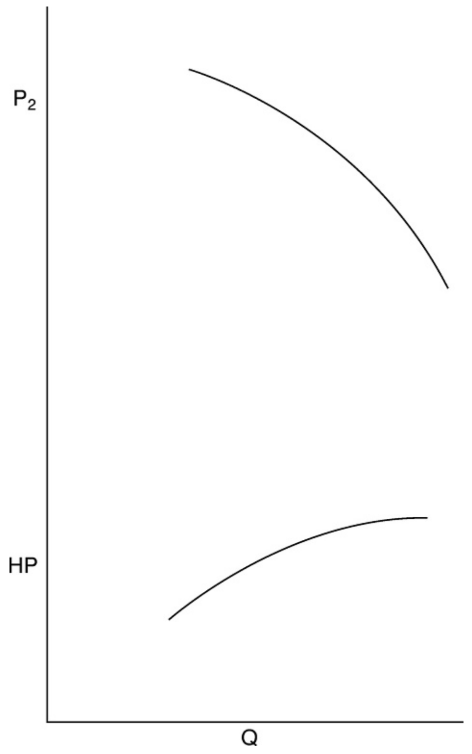


FIG. 6.3 Discharge pressure/horsepower performance curve.

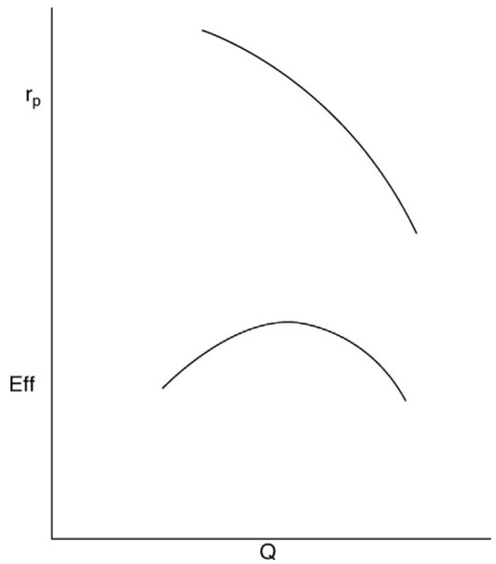


FIG. 6.4 Pressure ratio/efficiency curve.

START-UP[1]

Normal day-to-day operation of process compressors is relatively straightforward compared to the transient conditions of start-up and shut-down. During start-up, be it the first time or the 100th, things can change very quickly and it can be difficult to keep on top of the overall operation. This is one of the most crucial times in operating the equipment. The system

1. Adapted from "Compressor refresher," Elliott Company, Jeannette, PA, 1975, with permission [11].

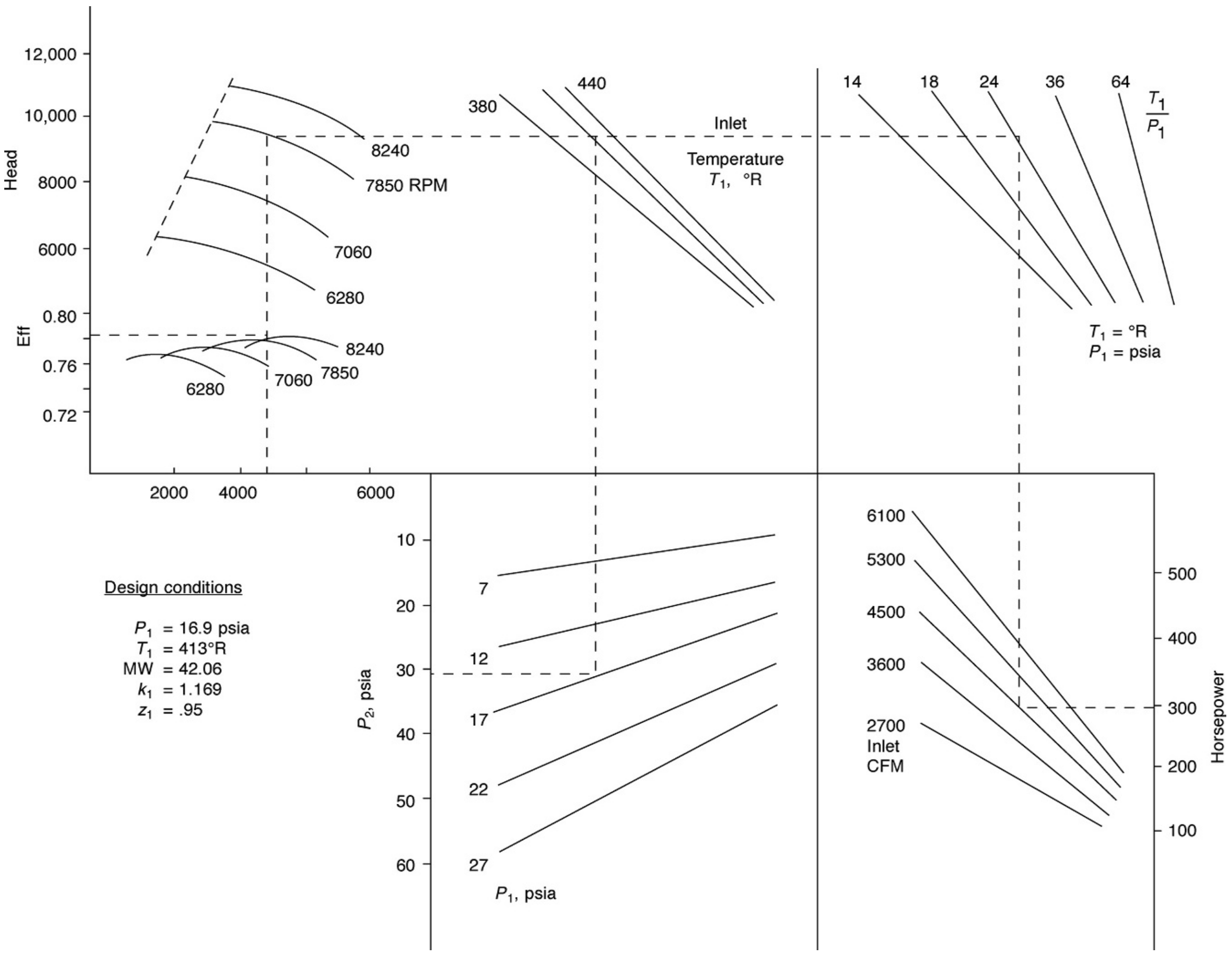


FIG. 6.5 Compressor performance in nomograph form.

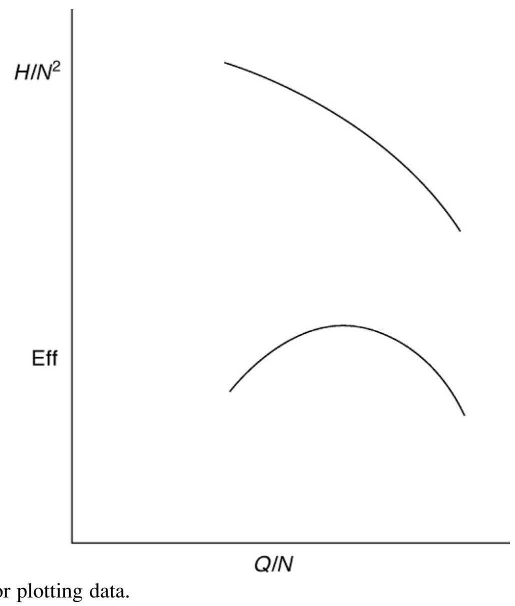


FIG. 6.6 Speed-compensated curve used for plotting data.

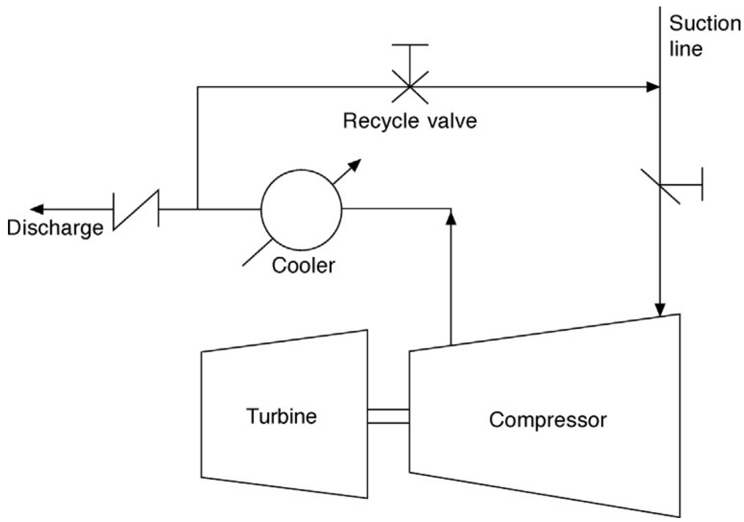


FIG. 6.7 Turbine-driven compressor.

must be designed, and the operators properly trained, to keep the equipment out of surge, not only during normal operation but also during start-up, shut-down, trip-out situations, or other process abnormalities.

The first start-up may last from several days to many weeks depending on the preparation and complexity of the system. Inlet screens, either temporary or permanent, should be installed on all compressor inlets to ensure that objects are not drawn into the compressor. Caution must be used, however, in the design of inlet screens so they cannot be drawn into the compressor or cause turbulence and affect the compressor performance. Install a $\triangle P$ gage across the screen to monitor the condition of the screen.

Occasionally, process compressors are operated on air to "dry out" the system before operation on normal process gas. Discharge temperature or seal operation may be the limiting factor here, but occasionally volume ratio effects may prevent operation. For best results, the manufacturer of the equipment should be consulted to review this off-design condition.

When starting a compressor on process, several different methods are available. The method used is a function of the system as well as the driver. For start-up of a steam or gas turbine-driven compressor such as shown in Fig. 6.7, coolant would be circulated through the cooler, the bypass or blow-off valve opened, and the unit brought up to operating speed.

The normal procedure for bringing a steam or gas turbine up to speed is usually governed by the manufacturer's recommended starting procedure for the turbine. This should be followed. Typically, this involves warmup of the turbine at 500–1000 RPM for a few minutes on a gas turbine to one-half hour or more on a steam turbine. Flow control on the compressor is not critical at this time as very little head or pressure is developed. After warming up, the unit's speed is gradually increased. Care must be exercised when approaching criticals, and they should be passed through rapidly. The ability to gradually increase speed is truly a plus feature of the turbine, as it provides the operators with time to make checks and adjustments as the speed is increased. Modern digital controls are a further plus as they can be programmed to not only control speed but also acceleration.

Fig. 6.8 shows a typical pressure ratio characteristic, which might be experienced as the steam turbine is brought up to speed. Take special precautions to avoid operation in the choke region for very long; otherwise, damage may occur to the compressor.

By providing a large bypass or recirculation system around the compressor, the pressure ratio across the unit can be kept low. This is especially valuable during turbine warmup.

Low pressure ratio and corresponding high flow mean very little chance for surge. This area, however, is an operating region where flow separation and damage to impellers can occur. On axial compressors, choke flutter can cause serious damage to the blading; therefore, operation of axial or centrifugal compressors in the overload (choke) regions should be avoided (see Figs. 4.10, 4.11, and 4.29–4.31).

Once the turbine is warmed up and the turbine speed ramp up begins, the bypass valve can be slowly closed, causing the compressor performance to ride up to the 100% speed line to design point "D" with precautions to keep the compressor away from choke.

If the compressor is to follow the path outlined here, the steam turbine must be capable of providing the required additional power. Point "A" (120% torque and 100% speed) corresponds to the peak horsepower typical for closed backward-leaning impellers. While 120% torque is not the worst case possible, it does represent a reasonably safe number for most applications.

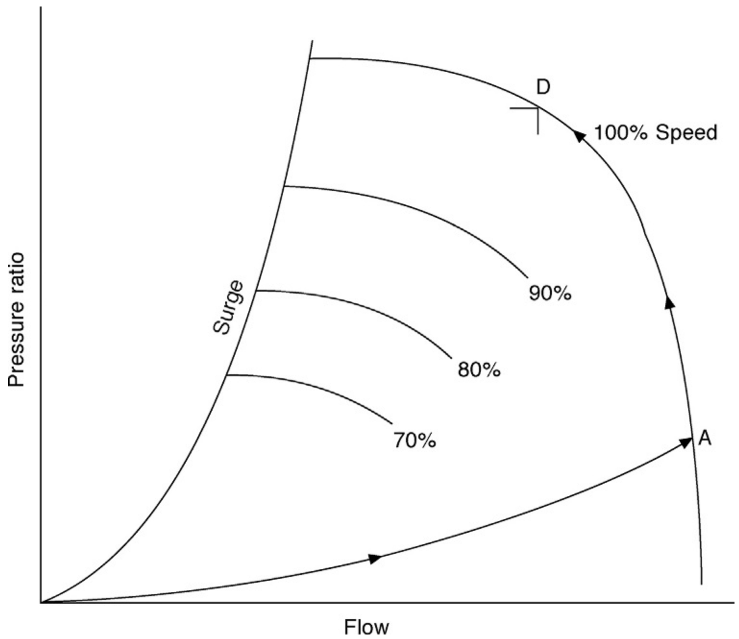


FIG. 6.8 Start-up of a turbine-driven compressor.

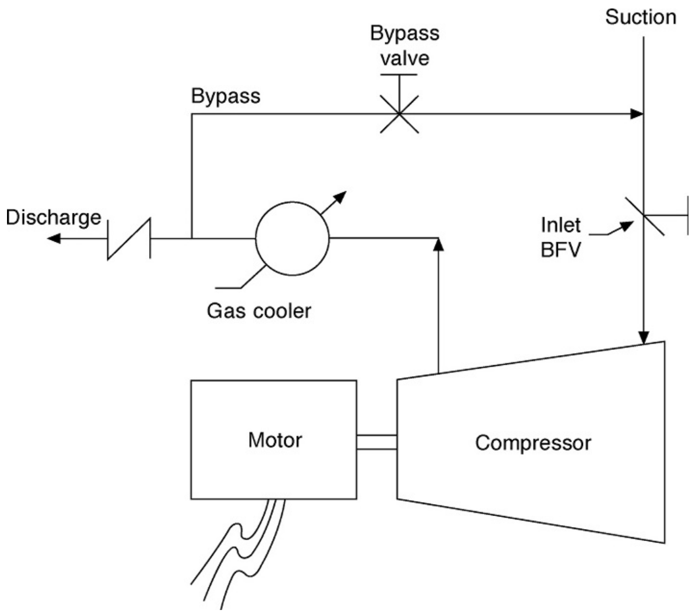


FIG. 6.9 Motor-driven compressor [11]. *(Used with permission of Elliott Company, Jeannette, PA.)*

Another type of system commonly encountered is the motor-driven centrifugal compressor. Unlike the steam turbine, the motor-driven unit usually has lower starting torque capabilities due to the need for limiting current inrush and heat buildup. Normal start-up time for a motor is limited to less than 20 s. Fig. 6.9 shows a typical simple motor-driven centrifugal compressor unit.

As in the steam turbine-driven unit, start the coolant circulating in the gas cooler and open the bypass valve prior to starting. In order to minimize current inrush, the inlet butterfly valve must be 95% closed. Typically, a mechanical stop is placed in the butterfly valve to assure the valve remains at least 5% open to assure the compressor does not go into surge during start-up. After this preparation, the machine should come to 100% speed within 10–20 s.

Preferably, the starting characteristic is such that the flow remains to the right of the surge line during the starting cycle (Point B, Fig. 6.10). When 100% speed is reached, the butterfly valve must be opened, bringing the compressor away from the surge area and further into the stable operating area.

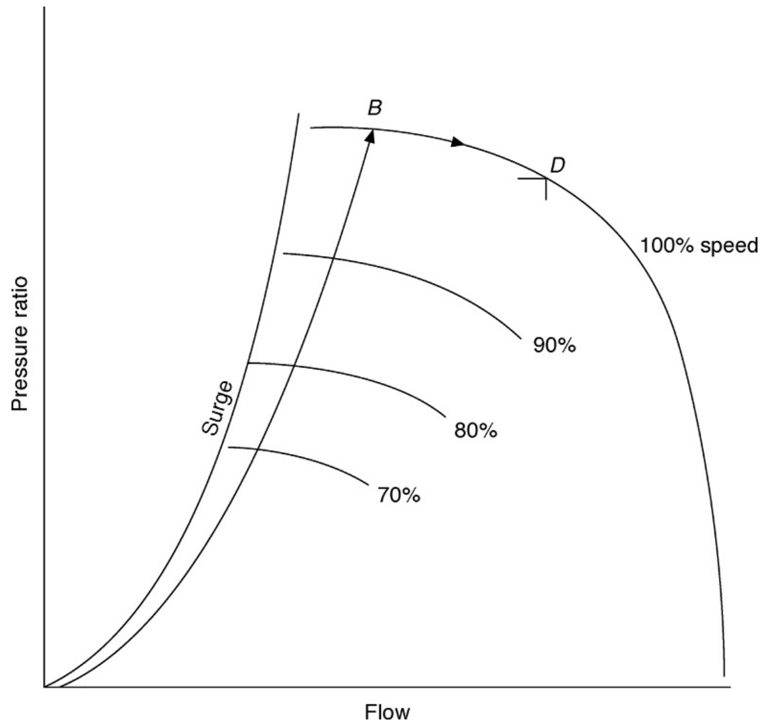


FIG. 6.10 Start-up of a motor-driven compressor.

As in the previous case with the steam turbine, the bypass valve should be wide open before starting the compressor. This puts the compressor discharge at or near suction pressure. In order to allow the compressor to develop its full design pressure ratio and keep it out of surge, the suction valve must be nearly closed to reduce the suction pressure. The reduced density will reduce the torque required on start-up and permit a smaller motor to be used (Fig. 6.11).

Once the compressor is up to speed, the inlet butterfly valve can be opened to increase suction density. The bypass valve can then be closed to raise the discharge pressure, and the suction valve can then be adjusted to maintain proper compressor loading.

On refrigeration units, the discharge flow normally goes to a condenser where the gas is condensed into liquid at a fixed pressure and temperature. In order to get the machine on stream, three things have to happen:

1. Inlet pressure must be kept low to provide low flash temperature.
2. At the same time, suction temperatures must be kept low so that pressure ratios can be realized.
3. Discharge pressure must be reached so condensing starts.

Thus, for a typical refrigeration system, once 30% speed is reached, the speed should be increased rapidly and at the same time the suction drums quenched with liquid in order to provide cooling. At the same time, gas may be blown off to keep suction pressures near design. As speed increases and inlet temperature falls, the blow-off can be closed and the gas diverted to the condenser. Bypasses must be controlled to maintain flows between surge and overload so that pressure ratios can be realized as design speed and temperature are approached.

On a compressor with sideloads or one that is intercooled, it is generally a good idea to establish flows and pressures across compressor sections starting at the suction and working toward the discharge.

MECHANICAL FIELD TESTS[2]

Frequently, for a new installation or major rebuild, a mechanical test is desired before start-up. Running such a test can prove out the new or rebuilt compressor along with the driver and auxiliaries before plant start-up. The desire is to find bugs in the system in time to minimize delays in plant start-up.

2. Adapted from "Guidelines for mechanical field testing of compressors," J. Dunaway, Elliott Company, Jeannette, PA, 1979, with permission [22].

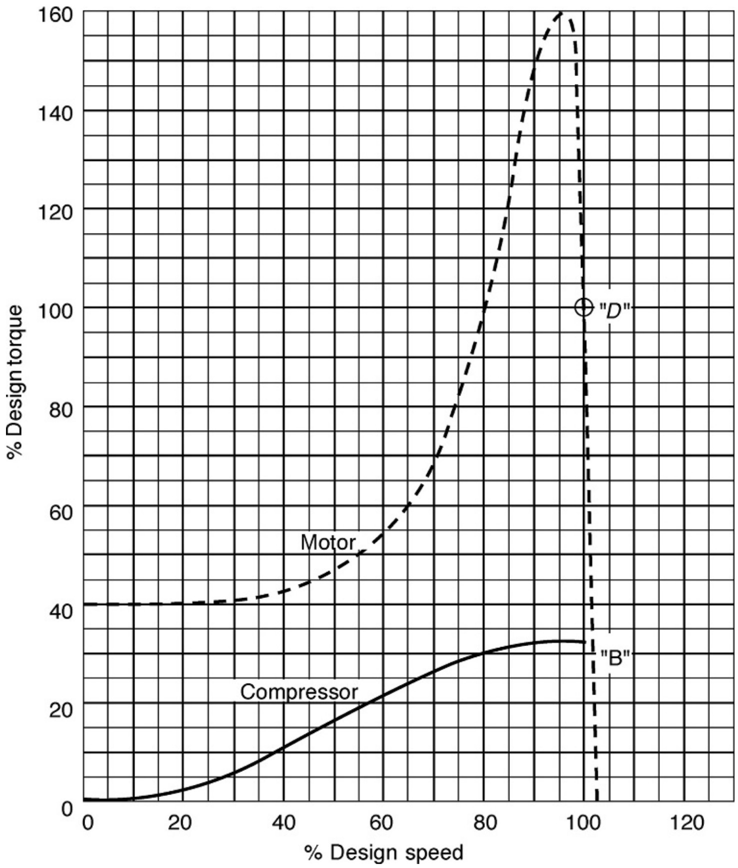


FIG. 6.11 Motor/compressor torque curve. Note that the motor torque is higher than the compressor torque. This difference provides the acceleration for start-up. Points B and D correspond to points B and D in Fig. 6.10. Torque at point B is low because the suction is throttled to provide low suction density and a corresponding low torque for start-up [11]. *(Used with permission of Elliott Company, Jeannette, PA.)*

Vacuum Tests

Running the compressor with the flanges blanked and 26–28 in. of Mercury vacuum pulled on the casing is a good way to prove mechanical integrity of the compressor string. This allows operation of the string to full speed yet with minimal horsepower requirements. Be sure to monitor the temperature at the discharge area, as it should not exceed the manufacturer's maximum temperature limit.

Although the compressor may be in surge during vacuum testing, there should be little temperature buildup and no mechanical damage due to the low energy levels in the gas.

If the unit has oil-lubricated seals, be sure to pull vacuum through the contaminated drains; otherwise, the contaminated oil will enter the compressor. Also, the seal oil ΔP may require hand control as the overhead tank or regulator may not properly maintain the ΔP. Check the seal bypass orifice or breakdown bushing clearance. These items may require adjustment. Due to the lower pressure, oil ΔP is reduced along with oil flow rates, which may cause overheating of the seals. Check with the manufacturer. Gas face seals may not fully obtain the proper lift at low pressure. Check with the seal supplier to be certain of seal capabilities.

One last thing on vacuum testing. The value of the testing is limited. Aerodynamic crosscoupling effects of internal labyrinth seals will not be demonstrated.

Open-Air Testing

An alternative to the vacuum test is an air test. Normally, the compressor is operated with a restricted inlet and an open discharge. As with any off-design operation, it is important to obtain advice from the OEM and stay within the limits specified. Pay particular attention to the discharge temperature and avoid the autoignition temperature shown in Fig. 6.12.

The suction opening can be an orifice plate with a "bird screen" over the opening to keep out debris. Size the orifice with the following procedure. Consider the suction orifice to control suction density, not flow. Assume the flow rate is controlled only by any discharge restriction (piping, check valve, elbows, etc.). With this in mind, and a known discharge pressure of

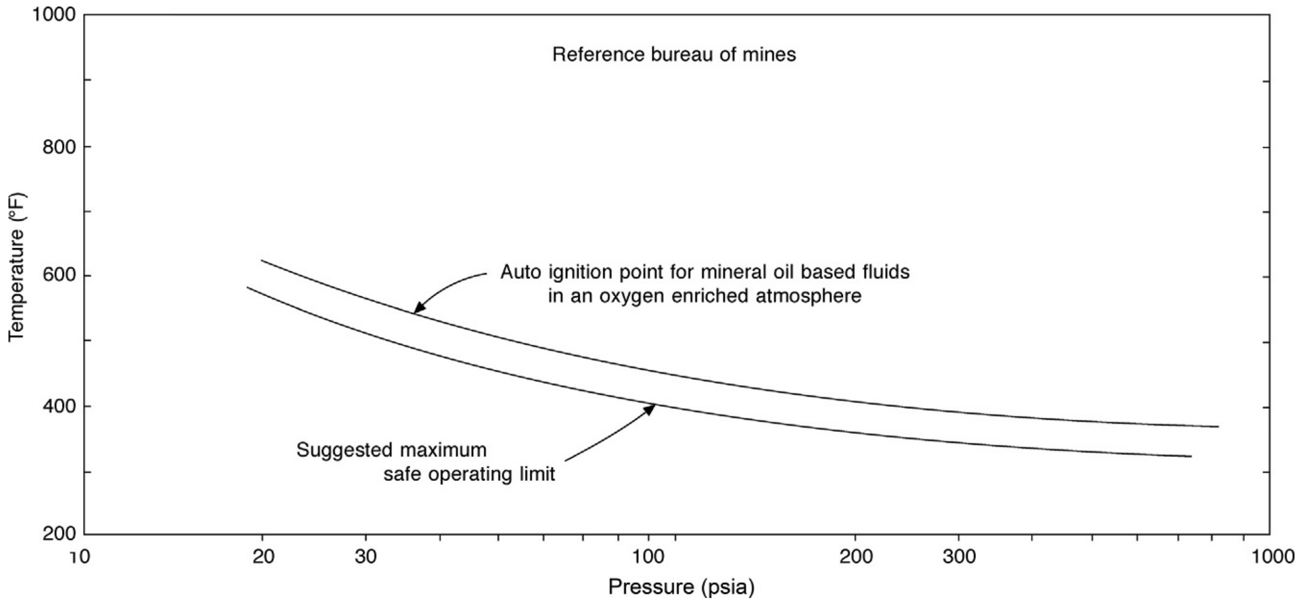


FIG. 6.12 Autoignition temperature. *(Data from Dunaway J. Guidelines for mechanical field testing of compressors. Jeannette, PA: Elliott Co.; 1979.)*

14.7 psia, assume normal design head and work back to obtain the suction pressure at the first impeller eye, using Eq. (7.4c). For example, if a suction pressure of 8 psia were calculated, the suction orifice would then be sized to pass design flow at 6.7 ΔP (14.7–8 psia).

This will assure that the compressor will operate near the design point at design head. Surge and/or high discharge temperature may, however, be unavoidable if the mole weight is significantly different than air [29] due to volume ratio effects (Fig. 4.18).

Note that as in the vacuum test, there may be some problems with the seals. Assure proper oil flow and watch for overheating of the seals. Buffer seals to minimize oil entering the machine.

Depending on the design density of the process gas, the air can result in power requirements above normal. If so, full speed may not be reached. Be sure to closely monitor the driver.

Verification of aeroinstability effects requires duplication of gas density. This may not occur for the air test described here.

One special note of caution on an air test. Too often, someone may desire to run an air test and simply remove the inlet and discharge piping and start up the unit on air. This is a big mistake! The compressor is attempting to make design head. Also, the pressure at the discharge is 0 (14.7 psia). This means that the first wheels are producing head, while the last wheel is turbinizing. Serious damage has occurred under this condition and it should be avoided.

Remember to allow the compressor to develop its design head. Any restriction at the discharge will limit flow, so this must be minimized. Reduce the suction density by throttling the inlet.

Full-Load Test

Sometimes, it may be possible to run a loop test in the field with normal process gas. If properly designed, the compressor will have a recycle loop capable of passing design flow. This loop will include a cooler, control valve, and orifice for measuring flow (Fig. 6.13). This situation will provide the best mechanical/aero test possible. All systems should be operating near normal conditions, thus closely simulating operation during plant operating conditions. Also, since process gas is being used, aerodynamic performance can also be verified.

AVOIDING SURGE[3]

The most important item for protecting the compressor from surge operation is the antisurge control system. This control system should maintain a minimum volume of flow through the machine so that the surge condition is never encountered.

3. Adapted from "Compressor performance," R. Salisbury, Elliott Company, Jeannette, PA, 1986, with permission [18].

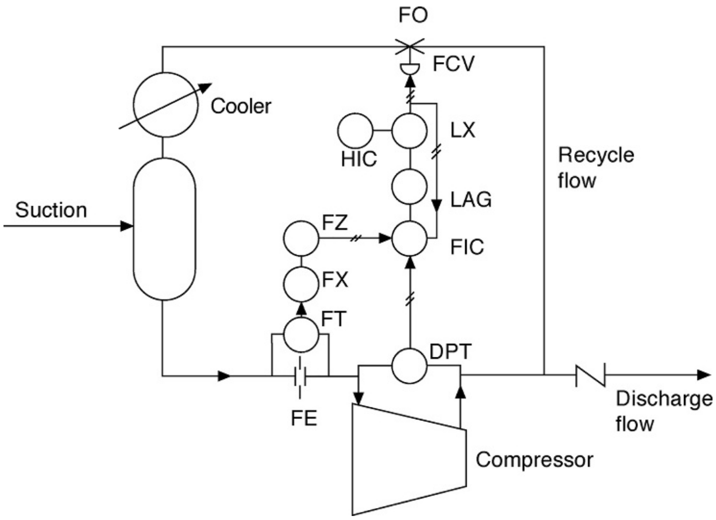


FIG. 6.13 Typical antisurge system [18]. *(Used with permission of Elliott Company, Jeannette, PA.)*

This is achieved by bleeding flow from the discharge of the machine to maintain a minimum inlet flow. This flow can either be dumped to atmosphere or recirculated back into the inlet of the compressor. In the latter case, it must be cooled to the normal inlet temperature. For most applications, a simple control based on a flow differential is adequate for this function. However, on machines where the speed or the gas conditions are variable, the control may have to be more sophisticated to insure proper operation under all conditions. This is frequently achieved by modulating the control with a signal for pressure, temperature, speed, or a combination of parameters.

Provisions must be made for start-up and trip-out of the machine with sufficient through flow to prevent surging and excessive heating of the inlet gas.

On any control scheme, a trip-out of the driver should be interlocked to open the antisurge valve within 3 s and allow the machine to coast to a stop with this line open. Otherwise, the machine could be surging constantly while dropping down in speed. This is particularly important for axial compressors and also for high-pressure, high-horsepower centrifugal applications. A typical antisurge control system is shown in Fig. 6.13.

A description of the function of each component is as follows:

- *FE* The flow element is usually an orifice located in the compressor suction, although it can be a venturi or calibrated inlet such as those used in axial compressors. Its purpose is to cause a temporary pressure drop in the flowing medium in order to determine the flow rate by measuring the difference of static pressures before and after the flow-measuring element.
- *FT* The flow transmitter is a differential pressure transmitter, which measures the pressure drop across the flow element and transmits a signal that is proportional to flow squared.
- *DPT* The differential pressure transmitter measures the differential pressure across the compressor and transmits an output signal that is proportional to the measured pressure differential.
- *FX* The ratio station receives the signal from the flow transmitter and multiplies the signal by a constant. This constant is the slope of the control line.
- *FZ* The bias station receives the signal from *FX,* the ratio station, and biases the surge control line.

The ratio station must have both ratio and bias adjustment to enable the control line to be placed as parallel to the compressor surge line as possible (see Fig. 6.14).

$$\Delta P = Ch + b \tag{6.1}$$

where ΔP is the calculated compressor differential signal; C is the control line slope (ratio signal); h is the inlet orifice differential signal measured by FT; and b is the control line bias (could be zero).

- *FIC* The surge controller is a flow control device, which compares the calculated output of FZ to the measured ΔP output of the DPT with ΔP as defined here.

When the calculated ΔP is greater than the measured ΔP, the compressor is operating to the right of the control line. When the calculated ΔP is equal to or less than the measured ΔP, the compressor is on or to the left of the control

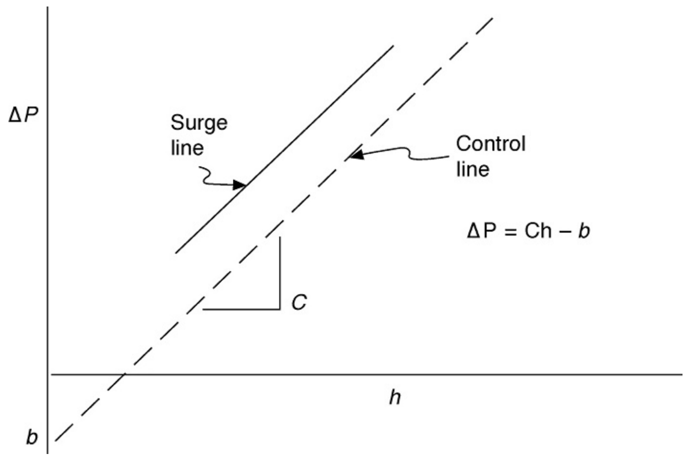


FIG. 6.14 Surge control line [18]. *(Used with permission of Elliott Company, Jeannette, PA.)*

line, and the surge controller functions as a flow controller and opens the antisurge valve as necessary to maintain operation of the compressor on this surge control line.

For rapid flow changes, the response of the control system must be rapid to prevent surge.

- *LAG* This device functions to enable the surge controller to open the recycle valve quickly, while providing a slow closure rate. This feature provides stability between the antisurge control system and the process by minimizing the hunting effect between control system and recycle valve.
- *LX* The low signal selector is set up for two inputs and one output. The inputs are a 100% signal valve and the surge controller output signal. The output of the low selector is sent to the recycle valve as well as back to the surge controller in the form of a feedback signal. This prevents the surge controller from winding up. Windup of the controller penalizes the reaction time of the antisurge control system.
- *FCV* The antisurge recycle valve functions to prevent surge by recycling flow from the compressor discharge back to the compressor inlet.

LIQUIDS[4]

One of the most potentially damaging occurrences for a compressor is the ingestion of liquid with the process gas stream. Liquids condensing in the recycle line can minimize the effectiveness of any antisurge system by creating a blockage in the line. The full range of operations should be studied to avoid having liquids enter the compressor during normal operation and upsets.

1. Trim cooling water or other process conditions to keep the compressor inlet conditions above the liquefaction points for any gas constituent.
2. Heat trace, bleed off, or purge normally stagnant lines when liquids form as the stagnant gas cools down to ambient temperatures.
3. Recycle lines should re-enter main gas stream either upstream of, or at the inlet knockout drum.
4. If any potential for liquid formation exists upstream or downstream of the compressor, drains and level indicators should be provided at all low spots of piping and vessels. This will allow routine checking for liquids and draining as required.
5. After shut-down, be sure all liquids formed by cooldown of the stagnant process gas are drained away before the compressor is restarted.

PARALLEL OPERATION

A common method of increasing capacity of a system while enhancing reliability is using two or more compressors operating in parallel. A combined characteristic of two identical units in parallel is shown in Fig. 6.15. Since the

4. Adapted from "Compressor refresher," Elliott Company, Jeannette, PA, 1975, with permission [11].

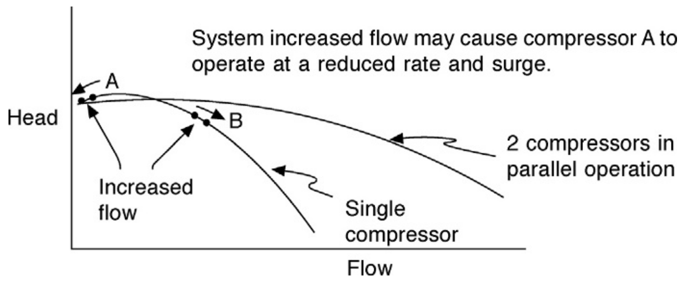


FIG. 6.15 Parallel operation. *(Data from Sheperd DG (Cornell University). Principles of turbomachinery. New York: Macmillan; 1956. p. 277.)*

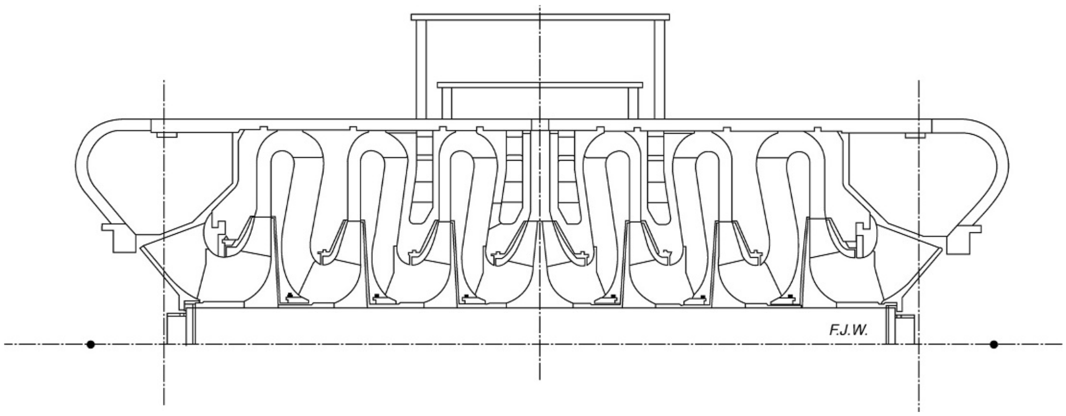


FIG. 6.16 Double-flow compressor.

"identical" units are always somewhat different and system resistance varies, it is feasible that both units will not be operating at the same point on the curve. It is possible for one unit to be in surge, while the other is not. It is therefore recommended that each unit have a separate antisurge system.

Double-Flow Compressors

In double-flow compressors (Fig. 6.16), a factor that must be considered is the design of the inlet piping to achieve a well-balanced, distortion-free flow into each inlet of the compressor. Otherwise, as with the parallel operation described, the compressor will not handle equal flow rates on each side and premature surging may occur before the antisurge control is activated.

INLET PIPING[5]

Compressor performance is very dependent on obtaining a uniform flow distribution to the impellers. Great pains are taken by the compressor designer to assure proper flow distribution to intermediate impellers. Although inlet guide vanes may exist on a compressor, this alone does not assure proper flow distribution to the first stage impeller. Compressors are designed assuming a relatively uniform velocity profile at the compressor inlet flange.

Although ASME goes to some detail in describing upstream straight-run requirements for orifice meters, the requirement for compressors is very simply stated. The straight-run requirement for axial inlet compressors is 10 pipe diameters and for nonaxial inlets, 3 diameters. Additionally, if the velocity pressure exceeds 1% of the static pressure, a flow equalizer must be used at the exit of the elbow upstream of the compressor inlet flange (Fig. 6.17).

5. Adapted from "Centrifugal compressor inlet piping—a practical guide," Ross Hackel and Raymond King, Elliott Company, Jeannette, PA, 1977, with permission [23].

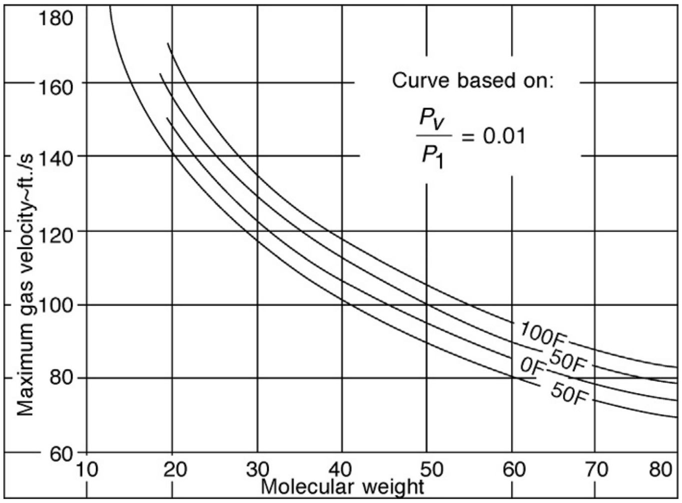


FIG. 6.17 Maximum velocity allowed at suction flange based on Eq. (6.2) [23]. *(Used with permission of Elliott Company, Jeannette, PA.)*

$$\frac{P_v}{P_1} \le 0.01 \tag{6.2}$$

where

$$P_v = \frac{P_1 V_1^2}{2gZRT_1} \tag{6.3}$$

Values for $P_v/P_1 = 0.01$ are shown in Fig. 6.17. Note that for 100°F air (MW = 29), the maximum inlet velocity for a three-diameter straight run without an equalizer is 140 fps. For propane (MW = 44) at 0°F, the maximum velocity would be 100 fps.

According to Hackel and King [23], the ASME guideline is adequate but could be modified to call for a reduced straight run for smaller P_v/P_1 values, and greater straight run for larger P_v/P_1 values. Also, additional length of straight run should be required for compound piping arrangements.

For a base case of one elbow in a plane parallel to the compressor axis and for a radial inlet with 50°F gas, Fig. 6.18 gives the minimum straight run required.

To correct for other suction temperatures, use the following equation to find the equivalent velocity for 50°F.

First calculate the actual velocity V_1.

$$V_{50F} = \frac{22.6 V_1}{\sqrt{T_1}} \quad \text{where } T_1 = {}^\circ R \tag{6.4}$$

For axial inlets and/or other inlet piping arrangements, use Fig. 6.18 along with the multipliers provided in Table 6.1.

Example 6.1

Consider a gas with a MW of 25, inlet temperature of 85°F, and suction velocity of 100 fps. The piping configuration is a radial inlet multistage compressor with two elbows in different planes upstream of the compressor. The elbow nearest the compressor is in a plane parallel to the rotor. Upstream of the two elbows is a butterfly valve.

Fig. 6.18 gives a straight-run requirement of 1.45 diameters. The multiplier for the two elbows is 1.75 and butterfly value factor is 3.0.

Multiplying

$1.45 \times 1.75 \times 3.0 = 7.6.$

Eight pipe diameters of straight run are required for this application.

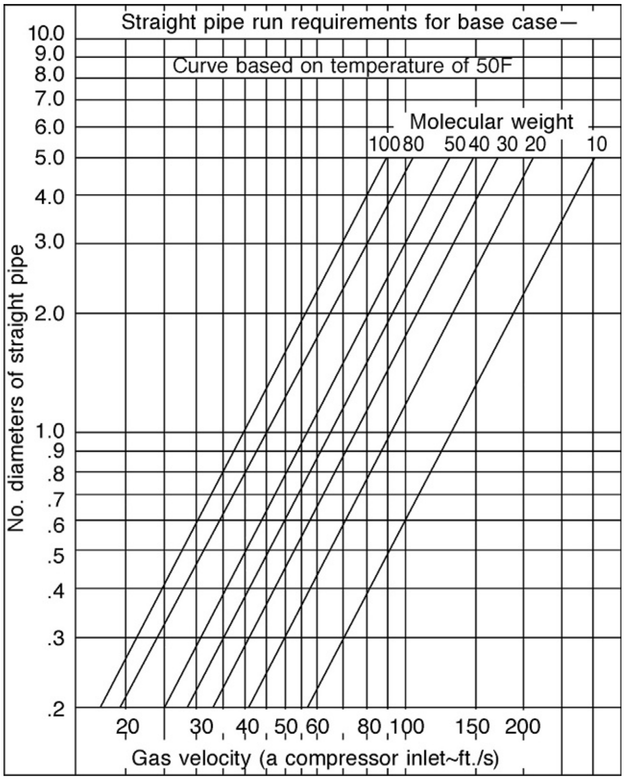


FIG. 6.18 Straight pipe run requirements for Case 1 in Table 6.1 [23]. *(Used with permission of Elliott Company, Jeannette, PA.)*

TABLE 6.1 Multipliers for Various Inlet Piping Arrangements[a]

Piping Configuration	Multiplier Radial Inlet
One long-radius elbow in a plane parallel to compressor shaft (Fig. 6.19)	1.0
One long-radius elbow in a plane 90° to compressor shaft	1.5
Two elbows at 90° to each other Elbow nearest compressor in a plane parallel to shaft	1.75
Above, only elbow normal to shaft	2.0
Butterfly valve before an elbow	
Valve axis normal to compressor shaft	1.5–3.0
Valve axis parallel to compressor shaft	2.0–4.0
Butterfly valve in straight run entering compressor inlet	
Valve axis normal to rotor	1.5–3.0
Valve axis parallel to rotor	2.0–4.0
Two elbows in the same plane parallel to compressor shaft	1.15
Two elbows in same plane 90° to rotor	1.75
Gate valve wide open	1.0
Swing check valve balanced	1.25
Axial inlet	1.25

[a]*Use with Eq. (6.4) and Fig. 6.18 to determine required straight run of piping for a given arrangement.*
From Hackel R, King R. Centrifugal compressor inlet piping—a practical guide. Jeannette, PA: Elliott Co.; 1977.

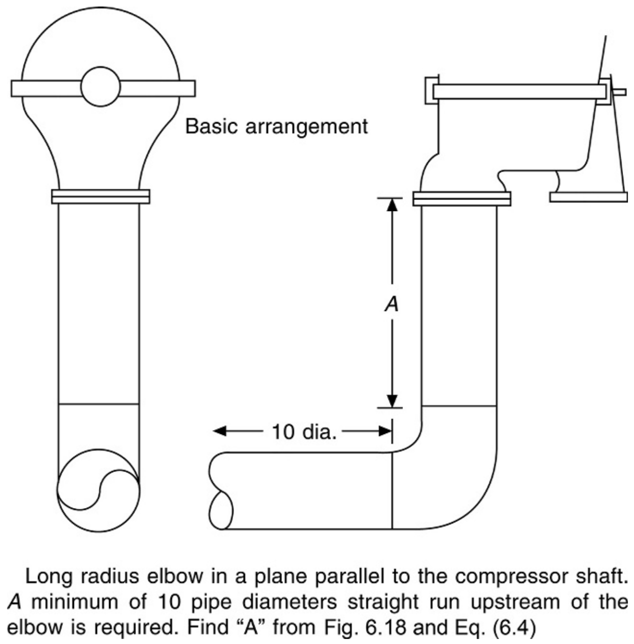


FIG. 6.19 Base case.

Flow Equalizer

When designing a flow equalizer, it is important to realize that pressure drop can be significant, which could do more harm than good, especially if the plate becomes plugged with debris. The best method for calculating pressure drop is to add the area of all the holes in the plate and determine an equivalent single hole orifice while calculating pressure drop accordingly. Be sure to note the effect of the recovery factor and install a delta pressure gauge so you can tell if the flow equalizer becomes plugged.

Mitered vaned elbows (Fig. 6.20) can provide an improvement over the perforated plate if the problem is solely from elbows. This type of elbow requires a minimum of space, has a very low loss factor, and generates a limited velocity profile

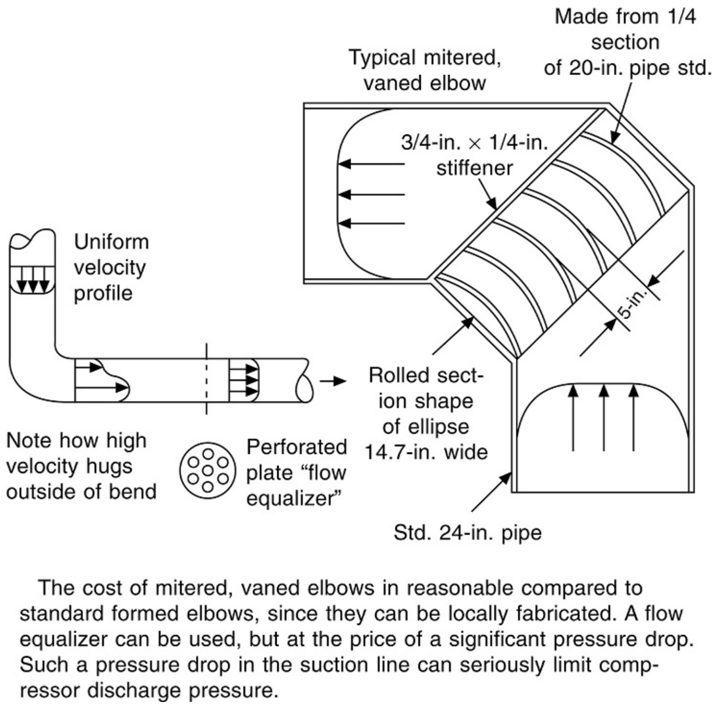


FIG. 6.20 Mitered vaned elbow for a 24″-diameter pipe.

distortion downstream of the elbow. Although a poor velocity profile will be carried through the elbow, two mitered vaned elbows in series at 90 degrees to each other will not create a downstream swirl.

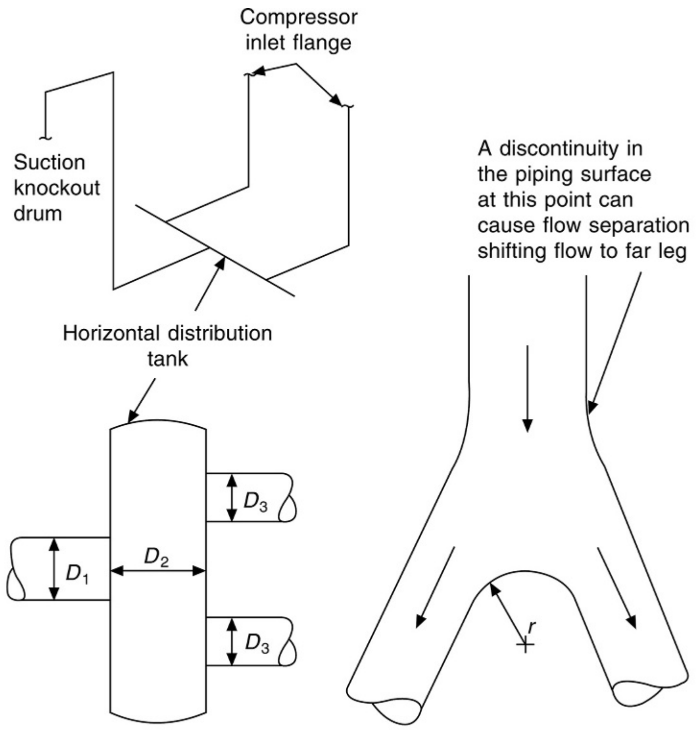
Double-Flow Compressors

A common method of increasing capacity of a system is using two or more compressors in parallel. However, since the "identical" units are always somewhat different and the piping is often not identical, both units will probably not be operating at the same point on the performance curve (Fig. 6.15). Therefore, it is always recommended that each unit have a separate antisurge system.

For a double-flow compressor, which is essentially two parallel compressors in a single body, application of separate antisurge systems is not practical. Therefore, it is crucial that the inlet piping be well designed to assure balanced flowrates to each section; otherwise, poor performance including premature surging will occur.

Experience has shown that the preferred arrangement uses a drum to split the flow (Fig. 6.21). While a Y with a proper upstream straight run of pipe may be an acceptable design, it should be noted that even a small discontinuity in the piping upstream of the Y can cause the flow to shift to one leg of the Y. A crucial factor affecting this phenomenon is velocity. Low velocity (relative to Fig. 6.17) will minimize this effect and result in improved flow distribution.

(Data from Compressor refresher. Jeannette, PA: Elliott Co.; 1975.)



The horizontal distribution tank shown on the left is the suggested piping for double flow compressors. D_3 and D_1 are sized according to Fig. 6.17. Size D_2 to achieve a velocity $^1/_4$ of that in Fig. 6.17. Note that the antisurge line should be fitted to the knockout drum further upstream and not to this distribution drum. Piping legs from the distribution drum to each inlet must be identical mirror image of each other.

A suggested design for a *Y* type splitter is shown on the right. Note the large radius at the dividing point. A mitered type joint with a sharp, pointed dividing geometry could cause flow separation and uneven distribution. A minimum of 10 pipe diameters is required upstream of a *Y* joint. Low velocity (relative to Fig. 6.17) will help assure equal flow distribution. This same *Y* joint geometry is suggested for rejoining the flow from the discharge of two parallel compressors.

FIG. 6.21 Piping for double-flow compressor and Y splitter.

Chapter 7

Field Performance Testing

This chapter provides a guideline for field performance testing of compressors. Included are procedures, equations, sample calculations, and data-trending procedures.

Before attempting a performance test, review the following checklist and be certain all the required data can be obtained. As a reference, see your instruction book for design conditions (see Appendix A, Table A.4).

- *Vane setting(s)*
- *Pressure and temperature at each flange*
- *Mass flow rate*
- *Gas properties*
- *Equipment speed*
- *Driver power*
- *Compressor and driver mechanical losses*

Field performance test procedures should be in accordance with ASME PTC10-1997 Compressors and Exhausters within practical limits. The method outlined here is for routine performance evaluation and is best if done on an automated, continuous basis to determine a long-term trend.

This procedure is not necessarily sufficient for an OEM acceptance test. If an acceptance test is to be performed, details should first be worked out with the manufacturer.

Compressor shaft horsepower is best determined by adding the enthalpy rise gas horsepower to standard values of bearing and seal mechanical power losses.

Mass flow is determined by using the process flowmeter. The flow rate is adjusted to account for variations from meter design flow conditions. Mass flow is checked by direct calculation from flowmeter upstream conditions and differential pressure using applicable ASME flow code equations.

Test point readings should not be taken until such time as the compressor is shown to be in equilibrium. Equilibrium is defined as the condition in which the discharge temperature does not vary more than 1°F over a 5-min period at constant inlet temperature. Upon achieving equilibrium, a snapshot of the data is taken and used for calculations.

It is recommended that a minimum of three and preferably five data points be taken if you are trying to establish the performance curve shape. Take one point at the design or normal operating point, at 95% of this value, 105%, 110%, and 90%.

A gas sample should be taken at the beginning and end of the test point. Gas sampling at the inlet can minimize condensation in the sample bottle. However, it is wise to sample at both suction and discharge. The recommended method of gas analysis is by gas chromatograph.

An estimate of overall test accuracy should be made by performing a power balance and/or noting the actual work input versus the predicted work input. Measured driver-delivered power minus gear (if applicable) power losses can be compared to calculated compressor-absorbed power. The overall accuracy of the test is no better than the accuracy indicated by the work input or power difference.

GAS SAMPLING

A stainless steel—sampling cylinder should be used. It should be at least 300 mL in size and have straight cylinder valves on both ends. The pressure rating of the cylinder should be high enough so that it can withstand full system pressure. To prevent gas condensation (the walls of the cylinder and tubing are relatively cool) during the sampling process, the cylinder and lines leading from the process pipe should be insulated and heat traced. The sampling container should be purged for about 5 min before closing the valves and trapping the gas at process pressure.

Compressor Performance. https://doi.org/10.1016/B978-0-12-814219-6.00007-6

The containers are then transported to the laboratory. During the transportation, the samples cool and therefore drop in pressure. The cooling of the sample may result in condensation of some of the gas. This condensation must be gasified before feeding into the gas chromatograph. The only sure way to do this is to heat the sample to or above the sample temperature. It may be wise to then bleed off some pressure before feeding the gas chromatograph. Plot this out on a Mollier diagram of the gas. This brings you further away from the dew point and provides further assurance of avoiding condensation.

A crosscheck on your accuracy can be made by checking the weight of a separate sample. This sample container should be at a vacuum and heated before filling with the process gas. Weigh the sample container evacuated and with the sample. Knowing the weight and volume will give you the specific volume. Check this against the calculated specific volume for temperature and pressure of the sample point using the composition given by the gas chromatograph.

Of course this procedure is not necessary in all cases, such as low-mole-weight mixtures. For heavy hydrocarbons, however, it is essential that this procedure be followed. Errors in gas analysis can give significant errors in performance.

INSTRUMENTATION[1]

General instrumentation requirements are shown in Fig. 7.1. All temperature pressure indicators must be dual, i.e., a minimum of two independent instruments per location.

Ideally, such as under development testing conditions, the static pressure tap hole should be very small (approximately 1/5) compared to the boundary layer thickness [25]. But on a more practical note, the static pressure connection should have a pressure tap hole .25 in. in diameter but no greater than 0.5 in., deburred and smooth on its inside edge, with a sharp corner. A smaller hole will collect dirt or condensate. Note that the hole must be deburred but have a sharp corner on the measurement side. Take care that the deburring procedure does not round the edges, as this will give erroneous readings (Fig. 7.2).

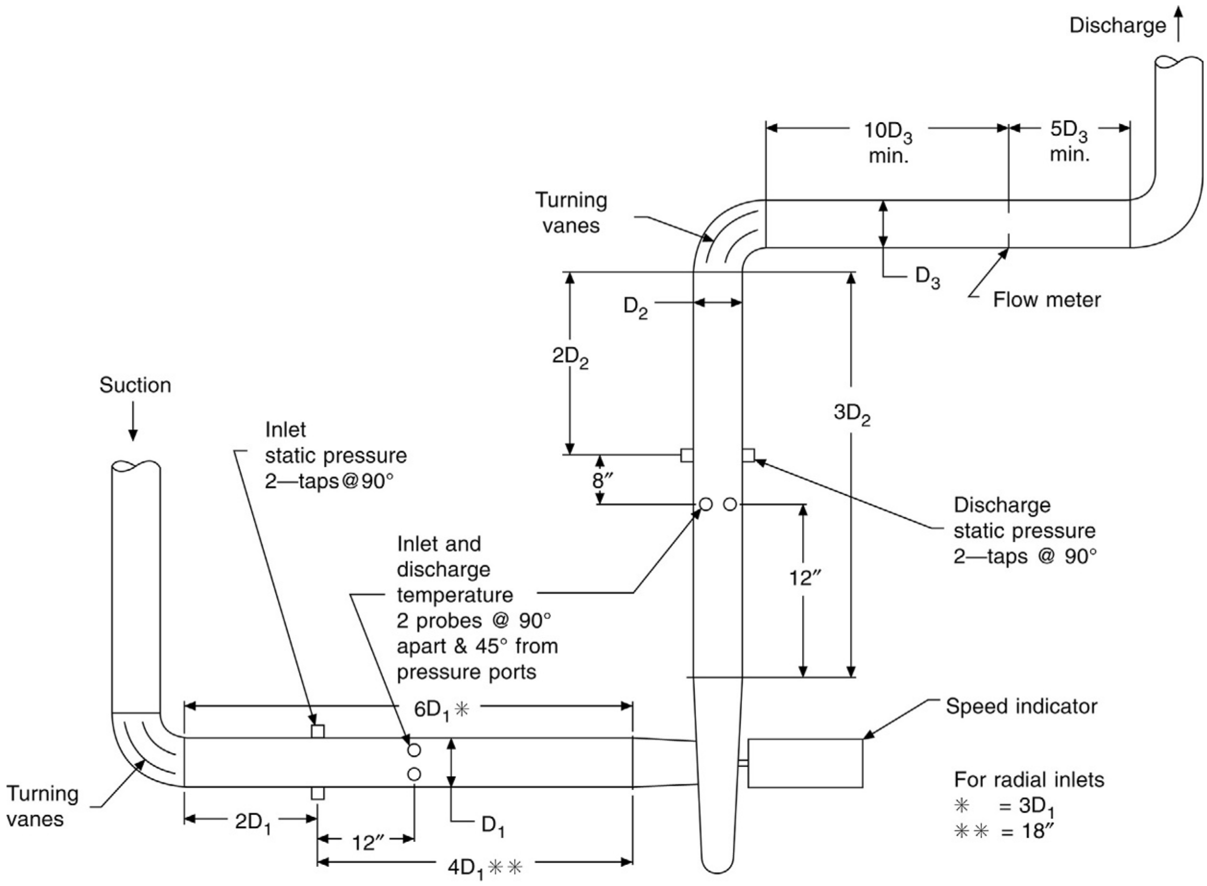


FIG. 7.1 Typical performance test setup.

1. Adapted from "Field performance testing," D. Bensema, Elliott Company, Jeannette, PA, 1986, with permission [24].

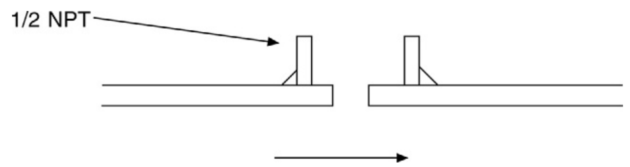


FIG. 7.2 Static pressure tap. The hole should be $\frac{1}{4}$ to $\frac{1}{2}$ in. in diameter. It should be deburred but have a sharp edge.

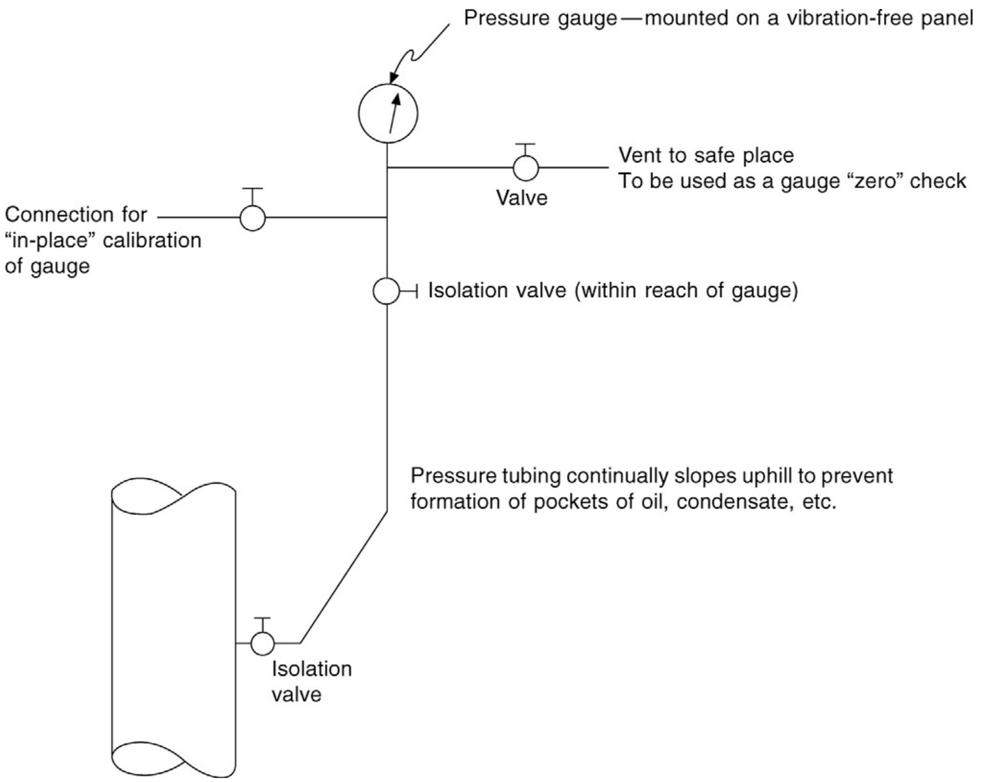


FIG. 7.3 Typical instrument line with gauges mounted above pressure tap. *(Data from Bensema D. Field performance testing. Jeannette, PA: Elliott Co.; 1986.)*

If possible, all pressures below 20 psig should be measured with a vertical-type fluid manometer of single- or double-leg design. Manometer fluids must be chemically stable when in contact with the test gases, and specific gravity must be measured before and after the test. If safety regulations do not permit the use of manometers, test gauges should be used.

All pressures above 20 psig at compressor flanges and flowmeter devices should be measured using quality gages, 6-inch or larger diameter, having a 0.25% sensitivity, and a maximum error of 0.5% full scale.

Pressure readings during testing should be at midscale or greater. Mounting should be on a vibration-free local panel, connected with pressure lines of at least 1/2 in. I.D. tubing; lines will continually slope down toward the unit to automatically drain any condensate. Block vent valves should be mounted at the gages to facilitate their in-place calibration. Calibration using a certified dead weight tester is preferred. Suggested arrangements are shown in Figs. 7.3 and 7.4.

Temperatures are to be measured using a thermocouple or RTD system having a sensitivity and readability of 0.5°F and an accuracy within 1°F. Care should be taken to avoid intermediate T-C junctions at terminals and switch boxes with a thermocouple system. Glass-stem thermometers are generally unacceptable for safety reasons. The temperature-sensing portion of the probe must be immersed into the flow to a depth of 1/3–1/2 the pipe diameter. The temperature-sensing elements should be in intimate thermal contact if using wells, utilizing a suitable heat transfer filling media, such as graphite paste. Stem conduction errors can be further minimized by wrapping the stem and well with fiberglass or wool insulation.

Speed should be determined utilizing two independent systems, one being the compressor key-phasor with calibrated digital readout with 0.25% or better system accuracy.

Pressure taps should be spaced 90° apart. On horizontal runs of pipe, pressure taps must be in the upper half of the pipe only.

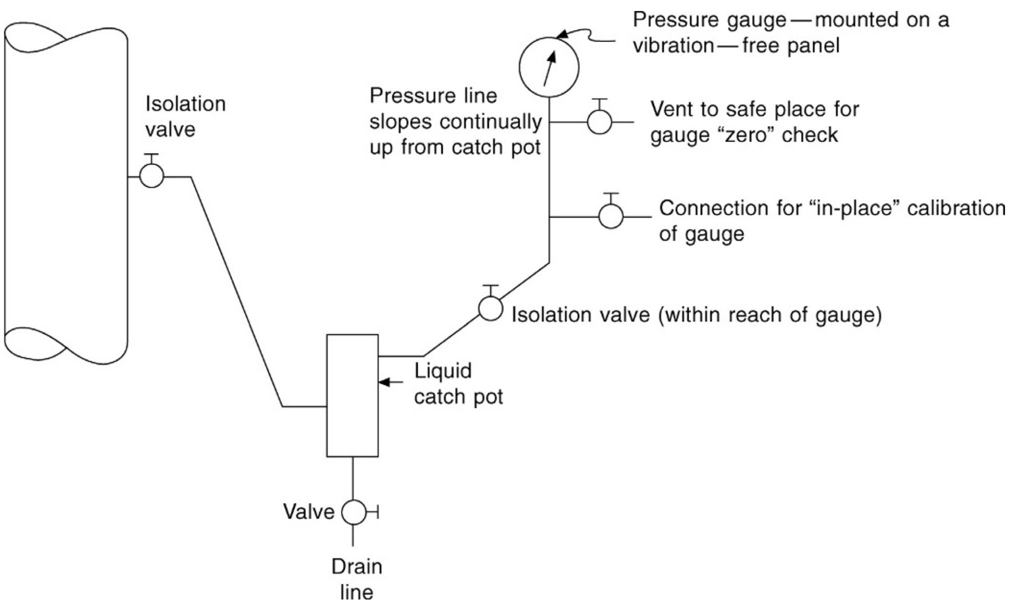


FIG. 7.4 Typical instrument line installation where pressure line cannot be sloped continuously upward from pressure tap to gauge. *(Data from Bensema D. Field performance testing. Jeannette, PA: Elliott Co.; 1986.)*

Flow rates derived from the process flow indicator should be checked by direct computation of mass flow rates through the metering device. For this reason, metering device upstream temperature, upstream pressure, and differential pressure must also be recorded.

Sideloads and Extractions

Preferred instrument locations are as shown in Fig. 7.5. Sideload and extraction lines, if applicable, are to be treated as inlet and discharge lines, respectively. If existing instrument tapping points must be used, care must be taken that those used are reasonably close to the compressor flanges. There must be no valves, strainers, silencers, or other sources of significant pressure drop between the pressure tap points and the compressor flanges.

Evaluation of sideload and extraction compressor performance is difficult and subject to significant inaccuracies unless internal pressure and temperature probes at the sideload and/or extraction lines are available.

Pressure and temperature taps can easily be added to a horizontally split compressor as shown in Fig. 7.5 by drilling and tapping the casing in the return channel crossover area. A minimum of two each pressure and temperature taps should be used. For barrel-type compressors, use the casing drain in this area if available. Although neither of these methods will provide precise data, they will obtain good relative data with which you can track the relative performance of your compressor. The data can become more realistic if a total mass balance and power balance is done on the system to determine accuracy of the test.

If data are required for an acceptance test, pressure and temperature probes should be located as shown in Fig. 7.5. Pressure may be calculated based on external flange measurements.

Once pressures and temperatures are known at the discharge of Section I, a mixing calculation is required to establish suction conditions for the next section (see Fig. 7.5).

$$P_1 = P_2 = P_3 \tag{7.1}$$

where 1 is the discharge of Section I; 2 is the sideload condition; and 3 is the mixed suction to Section II.

$$\dot{M}_1 h_1 + \dot{M}_2 h_2 = \dot{M}_3 h_3 \tag{7.2}$$

where $\dot{M}_1 + \dot{M}_2 = \dot{M}_3$.

T_3 is then found by working back through the Gas Properties or Mollier Diagram knowing h_3 and P_3.

T_3 may be very accurately approximated by

$$T_3 = \frac{\dot{M}_1 T_1 + \dot{M}_2 T_2}{\dot{M}_1 + \dot{M}_2} \tag{7.3}$$

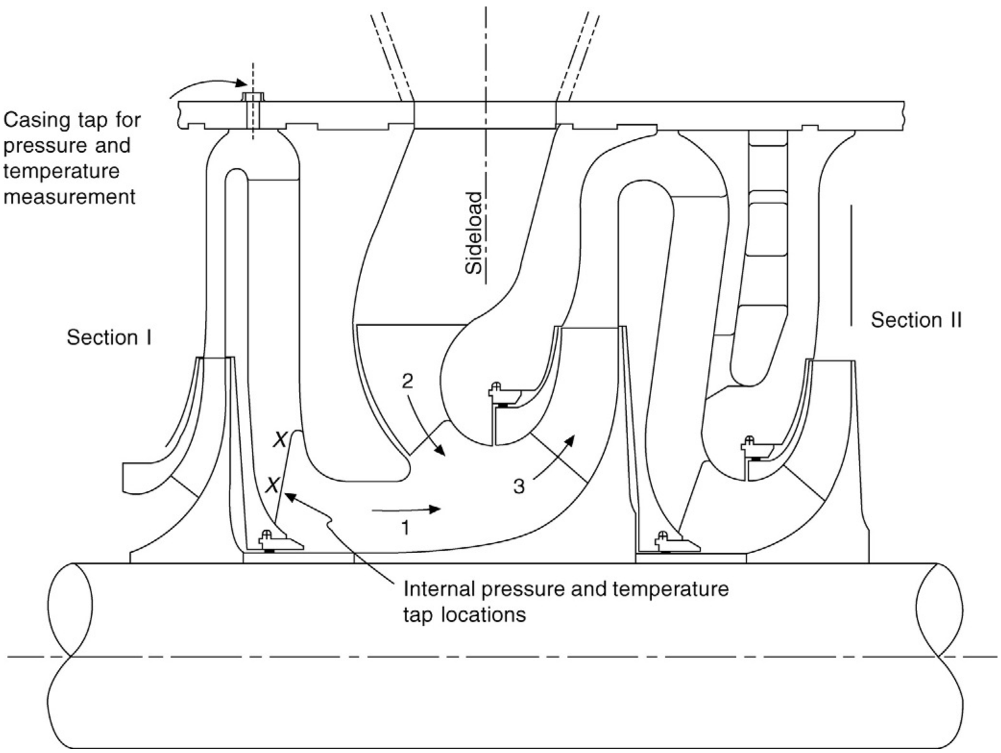


FIG. 7.5 Pressure and temperature taps for a sideload nozzle.

Special Data Reduction for Sideloads

Special data reduction techniques can be used on sideload and extraction compressors where internal pressures and temperatures are not available. Internal pressures can be estimated from flange pressures, gas velocity through the compressor nozzle, and standard pressure drop loss coefficients for a given sideload or extraction nozzle design.

Internal gas temperature at the discharge of each section is also required to determine sectional performance. This can be accomplished through an iterative process, which makes use of predicted work curves for each section. The procedure begins for a given test point by establishing the inlet volume flow for Section I. From the predicted work curves, the estimated work input is obtained. These data, along with the internal pressure determined earlier, are used to establish the estimated discharge temperature for Section I.

A BWR gas properties program like Gas Flex® is best used for this procedure (see Chapter 8). The sideload flow, as measured on site, will then be mixed with the calculated discharge flow from Section I to establish the inlet flow to Section II. This procedure is then repeated for each following compressor section using its respective work input plot. The test on the validity of the work input curves is made by comparing the calculated final discharge temperature with the measured final discharge temperature. If these two temperatures agree, the assumption is made that the correct work input has been used. If, however, the two temperatures do not duplicate one another within two degrees ($\pm 2°$), the work input curves for each section are varied by the same percentage, and the process is repeated.

Once the sectional inlet and discharge conditions are determined, the sectional heads and efficiencies can be calculated.

Instrument Calibration[2]

In general, all instruments used for the measurement of temperature, pressure, flow, and speed should be calibrated by comparison with appropriate standards before the test. General recommendations for calibration procedures are outlined later. A comprehensive log book should be maintained for all calibrations. Pressure gage calibration should state actual deadweight pressure and indicated gage value.

2. Adapted from "Field performance testing," D. Bensema, Elliott Company, Jeannette, PA, 1986, with permission [24].

Pressure gages should be check calibrated against deadweight standards throughout the range. Calibration using both increasing and decreasing pressure signals should be done to check for hysteresis. Gages not within 0.5% error of full scale should not be used. Needles should not be changed or adjusted. The gages must have a readable sensitivity to 0.25%.

There should also be a check on the accuracy of the thermocouple system (lead wires, reference junctions, readout devices) for each thermocouple. One method of accomplishing this is to read voltage output of the thermocouple locally, and then compare this to the remote reading of thermocouple output.

It is also recommended that the accuracy of the thermocouple itself be checked by subjecting it to varying temperatures and comparing its output to a reference standard. The thermocouple should be checked throughout its operating temperature range. The thermocouple system should have a readable sensitivity to 0.5°F and an accuracy within 1°F.

Calibration of the flowmeter differential pressure transmitter can be verified by impressing a known differential pressure across it and measuring its output. Finally, an overall system check can be made by impressing a differential pressure across the transmitter and reading the final control room output. The flowmetering device should be removed, and its dimensions should be checked, recorded, and compared to design criteria.

The tolerance for the measurement of compressor speed should not exceed 0.25%. Use of two independent instruments, one to provide a check on the other, is recommended. Electronic tachometers must be checked.

CALCULATION PROCEDURES[3]

General

Data reduction involves the following equations. Performance parameters are calculated using actual gas properties based upon the results of the on-site gas analysis. Performance parameters are calculated for each section of compression based on flange-to-flange data. The Mallen-Saville method used in Gas Flex is the preferred method for determining polytropic head and efficiency.

Head

1. Polytropic

$$H_p = Z_1 R T_1 \left(\frac{n}{n-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right] \tag{7.4a}$$

where

$$n = \frac{\ln(P_2/P_1)}{\ln(v_1/v_2)}$$

$$\frac{v_1}{v_2} = \frac{Z_1 R T_1/144 P_1}{Z_2 R T_2/144 P_2} = \frac{Z_1 T_1 P_2}{Z_2 T_2 P_1}$$

$$n = \frac{\ln(P_2/P_1)}{\ln(Z_1 T_1 P_2/Z_2 T_2 P_1)}$$

If head is known, and r_p or P_2 is desired,

$$\frac{P_2}{P_1} = \left[\frac{H_p}{Z_1 R T_1 [n/(n-1)]} + 1\right]^{n/(n-1)} \tag{7.4b}$$

$$P_2 = \left[\frac{H_p}{Z_1 R T_1 [n/(n-1)]} + 1\right]^{n/(n-1)} (P_1) \tag{7.4c}$$

3. Adapted from "Compressor performance," R. Salisbury, Elliott Company, Jeannette, PA, 1986, with permission [18], and "Field performance testing," D. Bensema, Elliott Company, Jeannette, PA, 1986, with permission [24].

Mallen-Saville equation (used in Gas Flex)

$$H_p = (h_2 - h_1) - \left[\frac{(s_2 - s_1)(T_2 - T_1)}{\ln(T_2/T_1)}\right] \tag{7.4d}$$

See also Eqs. (2.36), (2.37).

2. Adiabatic (Use for low-pressure air compressors.)

$$H_{\text{ad}} = RT_1\left(\frac{k}{k-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \tag{7.5a}$$

where H_{ad} is the adiabatic head, in feet.

$$k = c_p/c_v$$

$$\frac{P_2}{P_1} = \left[\frac{H_{\text{ad}}}{RT_1[k/(k-1)]} + 1\right]^{k/(k-1)} \tag{7.5b}$$

$$P_2 = \left[\frac{H_{ad}}{RT_1[k/(k-1)]} + 1\right]^{k/(k-1)}(P_1) \tag{7.5c}$$

See also Eq. (2.38).

Efficiency

1. Polytropic

$$\eta_p = \frac{H_p}{(h_2 - h_1)778.16} \tag{7.6}$$

where H_p is the polytropic head, in feet; h_2 is the enthalpy at discharge conditions in BTU/LB; h_1 is the enthalpy at inlet conditions in BTU/LB.

2. Polytropic—Constant k

$$\eta_p = \frac{(k-1)/k}{(n-1)/n} \tag{7.7}$$

3. Adiabatic (air compressors)

$$\eta_{\text{ad}} = \frac{T_1\left[(P_2/P_1)^{(k-1)/k} - 1\right]}{T_2 - T_1} \tag{7.8}$$

Flow

Basic flow measurement equations, ASME PTC 19.5-1971, "Fluid Meters." See also Chapter 10 for more information.

1. Square-edged Orifices

$$\dot{M} = (5.983)(K)\left(d^2\right)(\text{Fa})(Y)\sqrt{\frac{h_w}{v_1}} \tag{7.9a}$$

2. Flow nozzles and Venturi tubes

$$\dot{M} = (5.983)(C)(E)\left(d^2\right)(\text{Fa})(\text{Ya})\sqrt{\frac{h_w}{v_1}} \tag{7.9b}$$

Note: v_1 at meter conditions.

Work

$$\text{Work input, } W = \frac{H}{\eta_p} \tag{7.10a}$$

or

$$W = (h_2 - h_1)778.16 \tag{7.10b}$$

Gas Horsepower[4]

$$\text{GHP} = \frac{(h_2 - h_1)\dot{M}}{42.41} \tag{7.11}$$

or

1. Polytropic

$$\text{GHP}_p = \frac{H_p \dot{M}}{\eta_p 3300} \tag{7.12}$$

2. Adiabatic (air compressors)

$$\text{GHP}_{ad} = \frac{H_{ad} \times Q_1 \times 144 \times P_1}{\eta_{ad} \times T_1 \times R \times 33000} \tag{7.13}$$

$$\text{GHP}_{ad} = \frac{k}{k-1}\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \times \left(\frac{144 \times P_1}{\eta_{ad} \times 33000}\right) \times Q_1 \tag{7.14}$$

Shaft Horsepower

$$\begin{aligned} \text{SHP} &= \text{GHP} + \text{Bearing and Seal Horsepower} \\ &= \text{GHP} + (\text{GPM} \times \Delta T/12.6), \text{for light turbine oil} \end{aligned} \tag{7.15}$$

Reynolds Number

1. Pipe flow

$$\text{Re} = \frac{Vd}{\nu'} \tag{7.16}$$

where V is the gas velocity and d is the pipe diameter

2. Compressor Reynolds number

$$\text{Re} = \frac{Ub}{\nu'} \tag{7.17}$$

where U is the impeller or blade tip velocity, first stage (ft/s); b is the impeller tip opening, first stage (ft) centrifugal compressor (axial blade height at impeller outer diameter); and b is the Cord length, first stage (ft), axial compressor.

4. Casing heat transfer assumed to be zero.

Tip Velocity

$$\begin{aligned} U &= N\pi d/720 \\ &= Nd/229.3 \end{aligned} \tag{7.18}$$

where N is the speed; d is the tip diameter, in.

Specific Volume and Density

$$v = \frac{1}{\rho} = \frac{ZRT}{144P} \tag{7.19}$$

Acoustic Velocity

$$a = \sqrt{kg_c RT_1} \tag{7.20}$$

where k is the isentropic volume exponent (Eq. 2.24); g_c is the gravitational constant; R is the gas constant; and T_1 is the inlet temperature.

Mach Number

$$M = V/a \tag{7.21}$$

Viscosity

$$\begin{aligned} v' &= (\text{Viscosity in centipoise}) \times 6.72 \times 10^{-4} \times v \\ &= (\text{Viscosity in centistokes}) \times 1.076 \times 10^{-5} \\ &= \text{Kinematic viscosity, ft}^2/\text{s} \end{aligned} \tag{7.22}$$

Total Temperature

Temperature readings taken during a performance test are usually somewhere between static and total temperature. Static temperature is defined as the temperature that would be shown by a measuring instrument that has no relative velocity to the fluid stream being measured. In making performance calculations, it is necessary to consider the effects of velocity on the temperature readings. The stagnation or total temperature is defined as the temperature that would be measured at the stagnation point if the gas stream were brought to rest and its kinetic energy converted to an enthalpy rise by an isentropic compression process from the flow condition to the stagnation condition.

If using flange readings, it is not really necessary to be concerned with converting measured values to total temperature as the velocity is relatively low (less than 150 fps) in this area. If, however, readings are being taken in high-velocity areas (400 fps or more), such as the return channel area for sideload or extraction nozzles, total temperature is a must for obtaining accurate performance calculation results.

Static temperature readings can be converted to total temperature readings using the following equations.

$$T_{\text{total}} = T_{\text{static}} + T_{\text{velocity}}$$

where

$$T_{\text{velocity}} = \frac{V^2}{2g_C J c_P} \tag{7.23}$$

V is the fluid velocity, ft/s; T is the absolute temperature, °R; and J is the 778 ft-1b/Btu.

Measured temperature readings can be converted to total temperature readings using the following equation.

$$T_{\text{total}} = T_{\text{measured}} + (1-r)T_{\text{velocity}}$$

where r is the recovery factor (a value of 1–0).

Note: The factor 0.65 is an assumed velocity recovery factor for a plain sheathed thermocouple. Recovery factors for other styles of thermocouples may vary significantly.

Total Pressure

The normal pressure reading taken during a performance test is static pressure. Static pressure is the pressure in the gas measured in such a manner that no effect is produced by the velocity of the gas stream. However, in making performance calculations, a different pressure, total or stagnation pressure, is required. This pressure is the pressure that would be measured at the stagnation point when a moving gas stream is brought to rest and its kinetic energy is converted to an enthalpy rise by an isentropic compression from the flow condition to the stagnation condition. As with total temperature, conversion to total pressure is not always required. Only in high-velocity conditions, such as internal measurements for sideloads or extractions, is this procedure required.

The conversion from static to total pressure can be made as follows:

$$P_{\text{total}} = P_{\text{static}} + P_{\text{velocity}}$$

where

$$P_{\text{velocity}} = \frac{V^2}{2g_C 144v} \tag{7.24}$$

where V is the average fluid velocity $= Q/60A$; Q is the volume flow, CFM; A is the pipe area (ft^2); v is the local specific volume; and P is the absolute pressure, psia.

Abbreviated Parameters

In order to monitor the performance of a compressor, the best thing to do is to monitor head and efficiency versus flow as described in the previous section and compare predicted and previous test data on a continuous day-to-day basis. This will give you the up-to-date performance and trend information needed to predict when a maintenance shut-down is required for performance reasons and/or to help troubleshoot aerodynamic problems. Since a thorough test is not always practical on a continuous basis, some abbreviated parameters or methods of calculation are demonstrated here. When using these methods and procedures, remember that they are approximations and are not meant to replace the performance test described previously, but only to monitor trends.

Pressure ratio: A close look at the equations for calculating head (Eqs. 7.4a–7.4d, 7.5a–7.5c) will show that the primary variable in the equation is pressure, P_2 and P_1. Therefore, monitoring pressure ratio will give trends similar to monitoring head. Also, the plot of r_p versus Q will be similar to head versus Q.

$$r_p = \frac{P_2}{P_1} \tag{7.25}$$

Temperature rise: Compressor efficiency is defined as useful work done on the gas (head) divided by the total work. Since total work is directly related to enthalpy, which in turn is related to temperature, monitoring temperature rise will be an indication of total work input. If pressure ratio (r_p) goes down and/or temperature rise goes up (for a given flow and speed), then this is an indication that the efficiency of the compressor has gone down.

$$\Delta T = T_2 - T_1 \tag{7.26}$$

Flow: Orifice and nozzle meters: At many installations, flow is monitored in percent of some design amount:

$$Q = \left(\frac{\%}{100}\right) \times Q_D \tag{7.27}$$

where Q_d is the 100% design meter flow rate, SCFM and % is the meter reading, %.

Since this 100% flow was calculated for the design condition of the flow meter, the flow must be corrected for the actual conditions at the flow meter:

$$Q_c = Q \times \sqrt{\frac{P_A}{P_D} \times \frac{T_D}{T_A} \times \frac{MW_A}{MW_D} \times \frac{Z_D}{Z_A}} \tag{7.28}$$

where Q_c is the corrected flow, SCFM; A is the actual condition at flow meter; and D is the design condition of flow meter.

Also, the compressor characteristics are dependent on actual inlet flow, so the standardized flow must be corrected to inlet conditions.

$$Q_1 = Q_c \times \frac{P_S}{P_1} \times \frac{T_1}{T_S} \times \frac{Z_1}{Z_S} \tag{7.29}$$

where S is the standard conditions ($P = 14.7$ psia, $T = 60°\text{F}$, $Z = 1.0$) and 1 is the inlet conditions.

Flow: casing nozzle meter: Some equipment has built-in flow meters. Such is the case with the Elliott axial compressor. The flow in most compressors is generally accelerated somewhat before entering the first stage of compression. This can be equated to a Venturi effect and used to monitor flow rate.

In the case of the Elliott axial, each compressor inlet is calibrated during performance testing. The results are then plotted for various suction temperatures (see Fig.7.10).

TREND ANALYSIS

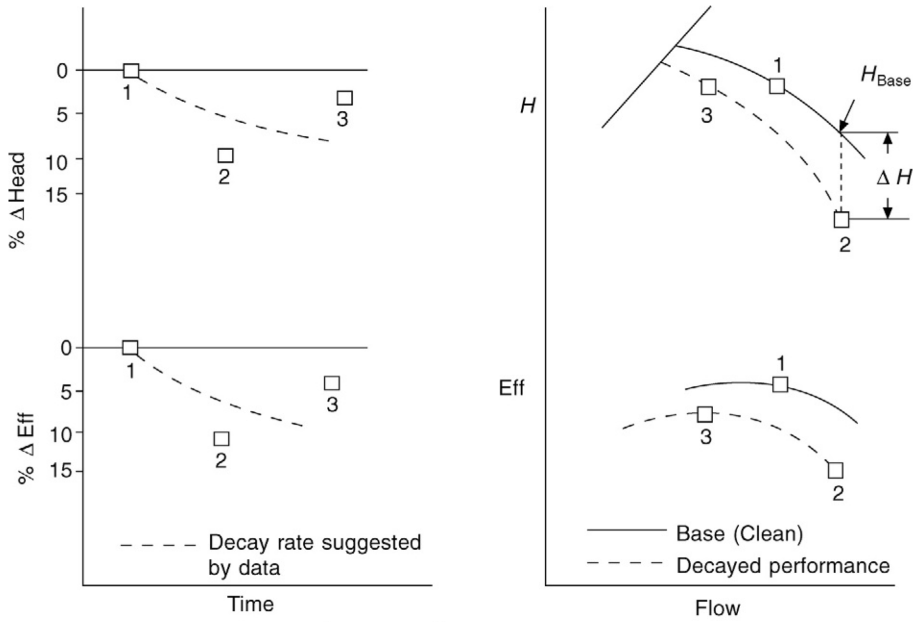
Accurate trend analysis on compressors can be somewhat confusing as the operating point and even the gas analysis may be continuously changing. Since this alone will affect the efficiency of the compressor, how can the trend be evaluated?

One method that has been used with great success is to plot percentage change of a parameter (efficiency, head and work input) to a known baseline (see Fig. 7.6).

It is important to first establish the baseline. Generally, the predicted performance curve is used. Preferably, the predicted curve is adjusted according to established field data for the compressor. Adjustments must be made for changes in inlet conditions, gas analysis, and speed. The test data are then compared to this "adjusted" prediction curve and the percent difference plotted versus time. Since performance degradation can be greater for off-design conditions, it is necessary to consider this effect when viewing the data. Some "educated guessing" is then required to determine how much the performance of the compressor has actually shifted. As shown in Fig. 7.6, the data can be somewhat deceiving; therefore, caution must be used in interpreting the data. Actual operating range will determine the urgency of any maintenance shut-down.

When compressor performance decays due to polymer buildup, dirt, corrosion, increased internal recirculation from seal wear, etc., the performance curve generally shifts downward and toward reduced flow as shown in Fig. 7.7.

Additionally, the process system may also be fouling. This means a greater restriction for a given flow. So, while the compressor has less head capability, more head may be required by the system.



$$\%\Delta H = \left(\frac{H_{\text{Base}} - H_2}{H_{\text{Base}}}\right) \times 100\%$$

FIG. 7.6 Trend analysis of a compressor performance. Note how the high-flow rate data (Point 2) suggest that the trend of decay in performance is rapid while data taken later at a lower flow rate (Point 3) show the decay to be at a lesser rate. Actually, there has been no decay in performance from Point 2 to Point 3.

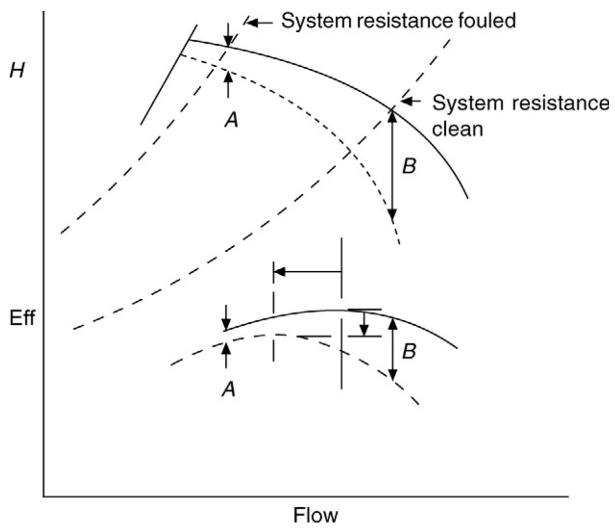


FIG. 7.7 Effect of fouling on compressor performance curves.

The efficiency is reduced because of the increased frictional losses and/or increased internal recirculation, shifting the performance down. This increased resistance also effectively reduces the capacity of the compressor, shifting the curves to the left. Even the shape of the curve will change some.

Continuous Monitoring

The problem in trending data can be minimized if a full range of continuous data is available over a long period of time. An example is shown in Fig. 7.8 where a compressor ingested a slug of rusty sludge. Fig. 7.9 shows the compressor after it was disassembled and cleaned. While this user selected to plot efficiency directly, delta efficiency (actual vs. predicted efficiency) can provide a clearer trend. Plotting delta head (actual vs. predicted head) will provide confirmation that the head is below prediction and plotting delta work input (actual vs. predicted work input) will confirm the accuracy of the data being analyzed.

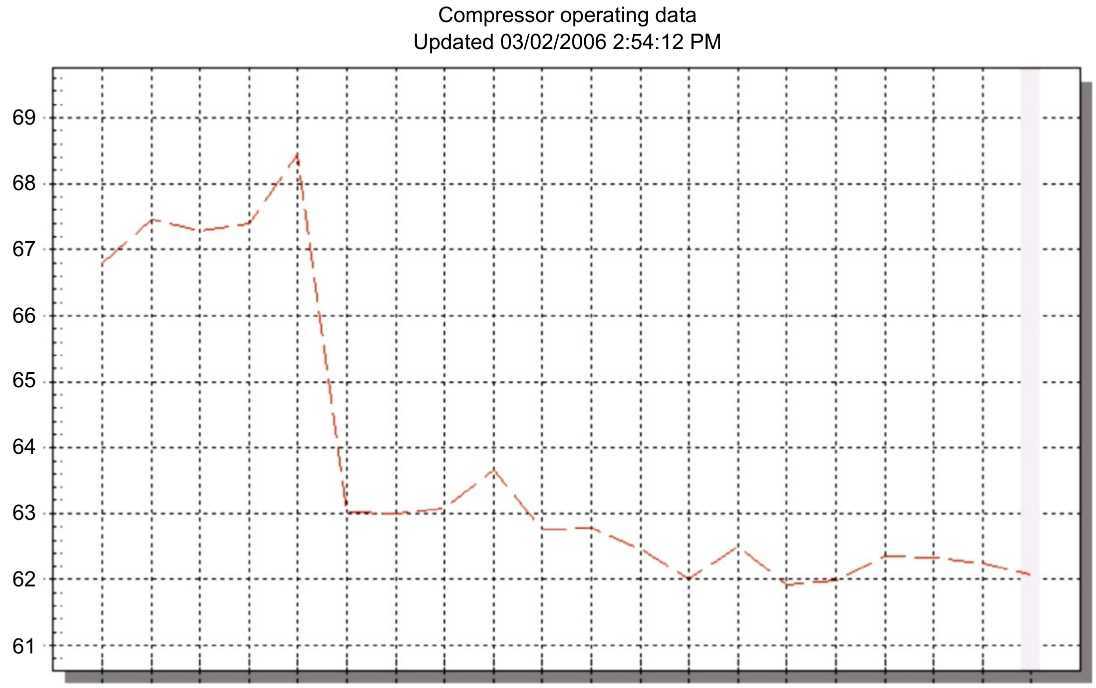


FIG. 7.8 Plot of efficiency versus time. Compressor ingested a slug of rusty sludge and efficiency dropped rapidly.

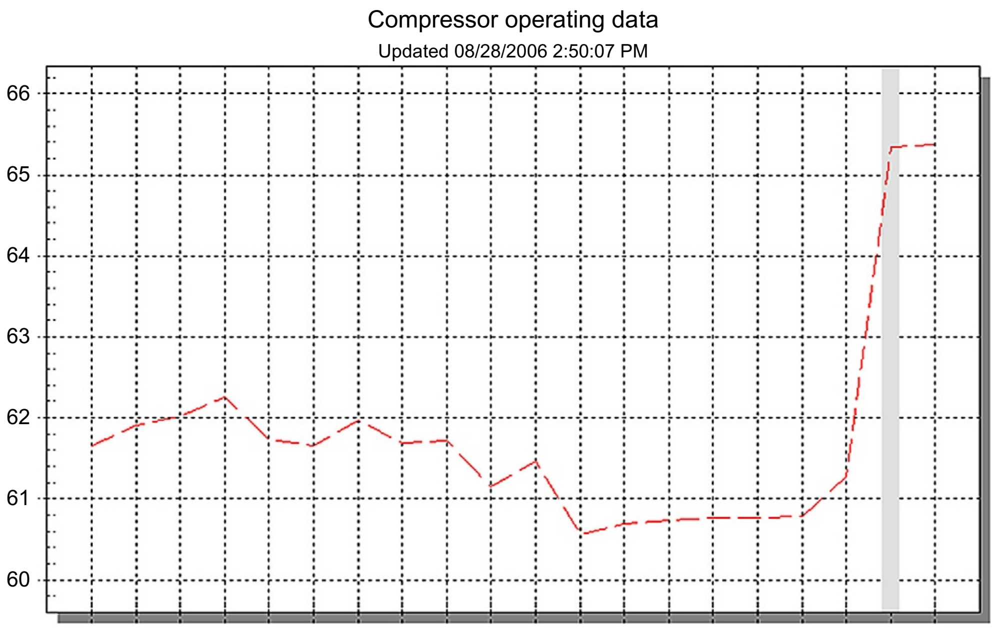


FIG. 7.9 After the compressor was opened, cleaned and the spare rotor installed, the efficiency returned to normal.

Sample Problem 7.1

Using Gas Properties Procedure
Gas analysis:

H_2	Hydrogen	34.12%
C_2H_4	Ethylene	33.57%
C_2H_6	Ethane	23.86%
CH_4	Methane	5.07%
CO	Carbon monoxide	3.38%

$P_1 = 233.4$ psig $\quad t_1 = 91.4°F$
$P_2 = 512.2$ psig $\quad t_2 = 257°F$
Speed = 6350 RPM
Atmospheric pressure = 14.5 psi
First convert data to the proper units.

$$P_1 = 233.4 \text{ psig} + 14.5 = 247.9 \text{ psia}$$
$$P_2 = 512.2 \text{ psig} + 14.5 = 526.7 \text{ psia}$$
$$T_1 = 91.4°F + 460 = 551.4°R$$
$$T_2 = 257°F + 460 = 717°R$$

From the gas properties program gas flex a BWR gas properties program
Inlet conditions

$P = 247.9$	$h = 220.3$
$Z = 0.9665$	$v = 1.212$
$T = 551.4$	$k = 1.265$
MW = 19.04	$c_p = 0.537$

Discharge conditions

$P = 526.7$	$h = 309.0$
$Z = 0.9803$	$v = 0.7524$
$T = 717.0$	$k = 1.240$
MW = 19.04	$c_p = 0.609$

Flow meter conditions

$P = 249.9$	$h = 220.4$
$Z = 0.9664$	$v = 1.203$
$T = 551.7$	$k = 1.265$
MW = 19.04	$cp = 0.538$

Absolute viscosity, $\mu' = 0.0148$ centipoise
Flow rate

Pipe diameter = 16″ OD, 15.25″ ID
Orifice diameter = 9.15″
$\beta = 9.15/15.25 = 0.6$
Orifice $\Delta P = 128.9$ in water

Calculate Reynolds number

$$\mathrm{Re} = \frac{Vd}{v'}$$

where

$$\begin{aligned} v' &= \text{Kinematic viscosity ft}^2/\text{s} \\ &= \text{Centipoise} \times 6.72 \times 10^{-4} \times \text{specific volume} \\ &= 0.01048 \times 6.72 \times 10^{-4} \times 1.203 \\ &= 8.47 \times 10^{-6} \end{aligned} \tag{7.22}$$

In order to calculate Reynolds number, the flow rate must first be known. For a start, use the design flow rate for the compressor. Assume $Q = 3600$ CFM:

$$\begin{aligned} V &= \frac{Q}{A} = \frac{3600\ \text{ft}^3}{\text{min}} \times \frac{4}{(3.14)(15.25\ \text{in.})^2} \\ &\times \frac{\text{min}}{60\ \text{s}} \times \frac{144\ \text{in.}^2}{\text{ft}^2} \\ &= 47\ \text{ft/s} \end{aligned}$$

$$\begin{aligned} \mathrm{Re} &= \frac{47\ \text{ft}}{\text{s}} \times \frac{\text{s}}{8.47 \times 10^{-6}\ \text{ft}^2} \times 15.25\ \text{in.} \times \frac{\text{ft}}{12\ \text{in.}} \\ &= 7.05 \times 10^6 \end{aligned}$$

From Chapter 10, Eq. (10.1):

$$\dot{M} = (5.983)(K)\left(d^2\right)(\mathrm{Fa})(Y)\sqrt{\frac{h_w}{v_1}}$$

$$K = \frac{C}{\sqrt{1-\beta^4}}$$

$$\dot{M} = 5.982\frac{CYd^2\mathrm{Fa}}{\sqrt{1-\beta^4}}\sqrt{\frac{h_w}{v_1}}$$

$C = 0.6049$ (from Table 10.2)
Fa = 1.000 (from Fig. 10.3)
$k = 1.265$

$x = \Delta P/P_1 = 128.9 \times 0.03613/250 = 0.0186$
$x/k = 0.0186/1.265 = 0.015$
$Y = 0.993$ from Fig. 10.4

$$\dot{M} = 5.982\left(\frac{0.6049 \times 0.993 \times 9.15^2 \times 1.00}{\sqrt{1-.60^4}}\right)\sqrt{\frac{128.9}{1.203}}$$
$$= 3337 \text{ lb/min}$$

At the compressor inlet:

$$Q = \dot{M} \times v_1$$
$$= 3337\frac{\text{lb}}{\text{min}} \times 1.212\frac{\text{ft}^3}{\text{lb}}$$
$$= 4044 \text{ ft}^3/\text{min}$$

Calculate the head

$$H = Z_1 R T_1\left(\frac{n}{n-1}\right)\left[(r_p)^{(n-1)/n} - 1\right] \tag{7.4a}$$

$$Z_1 = 0.966$$
$$R = 1545/\text{MW} = 1545/19.04 = 81.14$$
$$n = \frac{\ln(P_2/P_1)}{\ln(v_1/v_2)}$$
$$v_1 = 1.212\,\text{ft}^3/\text{lb}$$
$$v_2 = 0.7524\,\text{ft}^3/\text{lb}$$
$$n = \frac{\ln\,(526.7/247.9)}{\ln(1.212/0.7524)} = \frac{0.7536}{0.4768} = 1.581$$
$$r_p = \frac{P_2}{P_1} = \frac{526.7}{247.9} = 2.125$$
$$H = (0.966)(81.14)(551.4)\left(\frac{1.581}{0.581}\right)\left[2.125^{(0.581/1.581)} - 1\right]$$
$$H = 37538\,\text{ft-lb}_\text{f}/\text{lb}_\text{m}$$

Polytropic efficiency

$$\eta_p = \frac{H_p}{(h_2 - h_1)778.16}$$
$$= \frac{37538}{(309 - 220.3)778.16} \tag{7.6}$$
$$= 0.545$$

Gas horsepower

$$\text{GHP} = \frac{(h_2 - h_1)\dot{M}}{42.41}$$
$$= \frac{(309 - 220.3)3337}{42.41} \tag{7.11}$$
$$= 6979 \text{ HP}$$

Power balance
Motor power

$$\text{BHP} = E \times I \times \eta \times \text{P.F.} \times \sqrt{3}/746$$
$$= 6600 \times 444 \times 0.96 \times 0.92 \times 1.732/746$$
$$= 6009\,\text{HP}$$

Mechanical losses

Mechanical losses are generally a small percentage of the overall horsepower; therefore, exact numbers are not critical to the overall outcome. If the values are significant, then the losses can be verified by measuring oil flow rates and temperature rise. For light turbine oil (32 SSU), HP = oil flow rate in gpm × ΔT/12.6.

57 compressor bearings and seals
21 motor bearings
85 gear

163 total

$$\text{Test Error} = \left[\left(\frac{\text{Driver Power} - \text{Losses}}{\text{GHP}}\right) - 1\right] \times 100\%$$
$$= \left[\left(\frac{6009 - 163}{6979}\right) - 1\right] \times 100\%$$
$$= 16.2\%\text{Error}$$

Note that compressor power is higher than driver power and efficiency is low! This indicates something is probably wrong with the performance test. Recheck driver power, gas properties, pressure measurement method, etc.

Example Performance Test 7.2

Axial Air Compressor

Test data:

$P_1 = 14.5$ psia	$T_1 = 56°F = 516°R$
$P_2 = 54.5$ psia	$T_2 = 349°F = 809°R$
Inlet $\Delta P = 40$ in. H_2O	$N = 5804$ RPM

Head

$$H_{ad} = RT_1\left(\frac{k}{k-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right]$$
$$= \left(\frac{1545}{28.97}\right) \times 516\left(\frac{1.4}{1.4-1}\right)\left[\left(\frac{54.5}{14.5}\right)^{(1.4-1)/1.4} - 1\right] \tag{7.5a}$$
$$= 53.33 \times 516 \times 3.5\left(3.76^{0.286} - 1\right)$$
$$= 44,353\,\text{ft-lb/lb}$$

Efficiency

$$\eta_{ad} = \frac{T_1\left[(P_2/P_1)^{(k-1)/k} - 1\right]}{T_2 - T_1}$$
$$= \frac{516\left[(54.5/14.5)^{(1.4-1)/1.4} - 1\right]}{809 - 516} \tag{7.8}$$
$$= \frac{516 \times 0.46}{293}$$
$$= 0.81$$

Horsepower

Inlet flow for this compressor is easy to check since it has a calibrated inlet. From calibration curves, Fig. 7.10,

$$Q_1 = 92,000\ \text{ICFM}$$
$$\text{GHP} = \frac{H_{ad} \times Q \times 144 \times P_1}{\eta_{ad} \times T_1 \times R \times 33,000}$$
$$= \frac{44,353 \times 92,000 \times 144 \times 14.5}{0.81 \times 516 \times 53.33 \times 33,000} \tag{7.13}$$
$$= 11,583$$

Power balance

Motor power

$$\text{BHP} = E \times I \times \eta \times \text{P.F.} \times 1.732/746$$
$$= 13,800 \times 442 \times 0.95 \times 0.91 \times 1.732/746$$
$$= 12,244\,\text{HP}$$

Mechanical losses

75 compressor bearings
35 motor bearings
235 gear

345 total

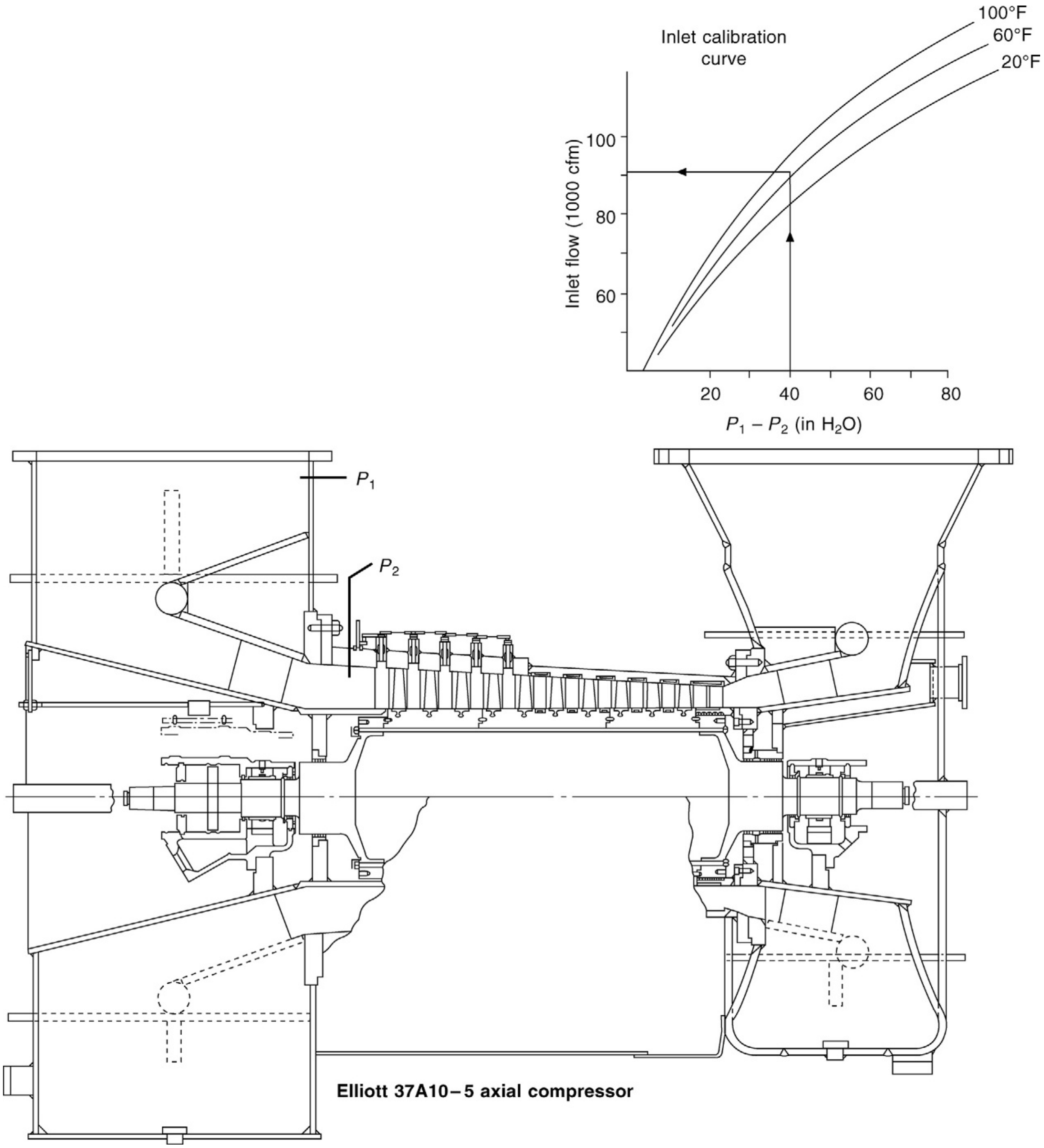


FIG. 7.10 Inlet calibration curve.

$$\text{Test Error} = \left[\left(\frac{\text{Driver Power} - \text{Losses}}{\text{GHP}}\right) - 1\right] \times 100\%$$

$$= \left[\left(\frac{12,244 - 345}{11588}\right) - 1\right] \times 100\%$$

$$= 2.7\%$$

Example 7.3

Mollier Method

Note that Mollier diagrams are available for almost any gas or gas mixture; several are shown in Appendix B. While it is much easier and more accurate to use a gas properties program like Gas Flex, this method of using Mollier diagrams was popular prior to the availability of laptop computers.

Gas mixture: 100% propane

$P_1 = 14.5$ psia	$T_1 = 40°F, 500°R$
$P_2 = 305$ psia	$T_2 = 312°F, 772°R$

Flow meter, 85%

Flow meter design conditions

$Q = 6.0$ MMSCFD (million standard cubic feet per day)

$P = 16.7$ psia MW = 44

$T = 50°F$ $Z = 0.977$

Gas properties from Mollier diagram, Fig. 7.11

$v_1 = 8.4\ \text{ft}^3/\text{lb}$ $v_2 = 0.57\ \text{ft}^3/\text{lb}$

$h_1 = 128$ BTU/lb $h_2 = 242$ BTU/lb

$Pv = ZRT$

$$Z = \frac{Pv}{RT}, R = \frac{1545}{44} = 35\frac{\text{ft-lb}_f}{\text{lb}_m\text{-}^\circ\text{R}}$$

$$Z_1 = \frac{14.5 \times 8.4 \times 144}{35 \times 500}$$

$$= 1.00$$

Flow rate

$$Q = \frac{85\%}{100} \times 6.0\,\text{MMSCFD}$$

$$= 5.1\ \text{MMSCFD}$$

$$= \frac{5.1\,\text{MM STD.FT}^3}{\text{DAY}} \times \frac{10^6\,\text{STD.FT}^3}{\text{MM STD.FT}^3}$$

$$\times \frac{\text{DAY}}{24\,\text{HRS}} \times \frac{\text{HR}}{60\,\text{MIN}}$$

$$= 3542\ \text{SCFM (standard cubic feet per minute)}$$

Using Eq. (7.28):

$$Q_c = Q\sqrt{\frac{P_A}{P_D} \times \frac{T_D}{T_A} \times \frac{MW_A}{MW_D} \times \frac{Z_D}{Z_A}}$$

$$= 3542\sqrt{\frac{14.5}{16.7} \times \frac{510}{500} \times 1.0 \times \frac{0.977}{1.0}}$$

$$= 3294\,\text{SCFM}$$

Using Eq. (7.29):

$$Q_1 = Q_c \times \frac{P_S}{P_1} \times \frac{T_1}{T_S} \times \frac{Z_1}{Z_S}$$

$$= 3294\left(\frac{14.7}{14.5} \times \frac{500}{520} \times \frac{1.0}{1.0}\right)$$

$$= 3212\,\text{ICFM (inlet cubic feet per minute)}$$

Head

$$H_p = 72\left[\ln\left(\frac{P_2}{P_1}\right)\right](P_1 v_1 + P_2 v_2) \tag{2.36}$$

$$= 72\left[\ln\left(\frac{305}{14.5}\right)\right](14.5 \times 8.4 + 305 \times 0.57)$$

$$= 64,800 \qquad (\text{Ref. } 63,966)$$[5]

5. Reference numbers indicate values obtained using the gas properties procedure demonstrated in Sample Problem 7.1.

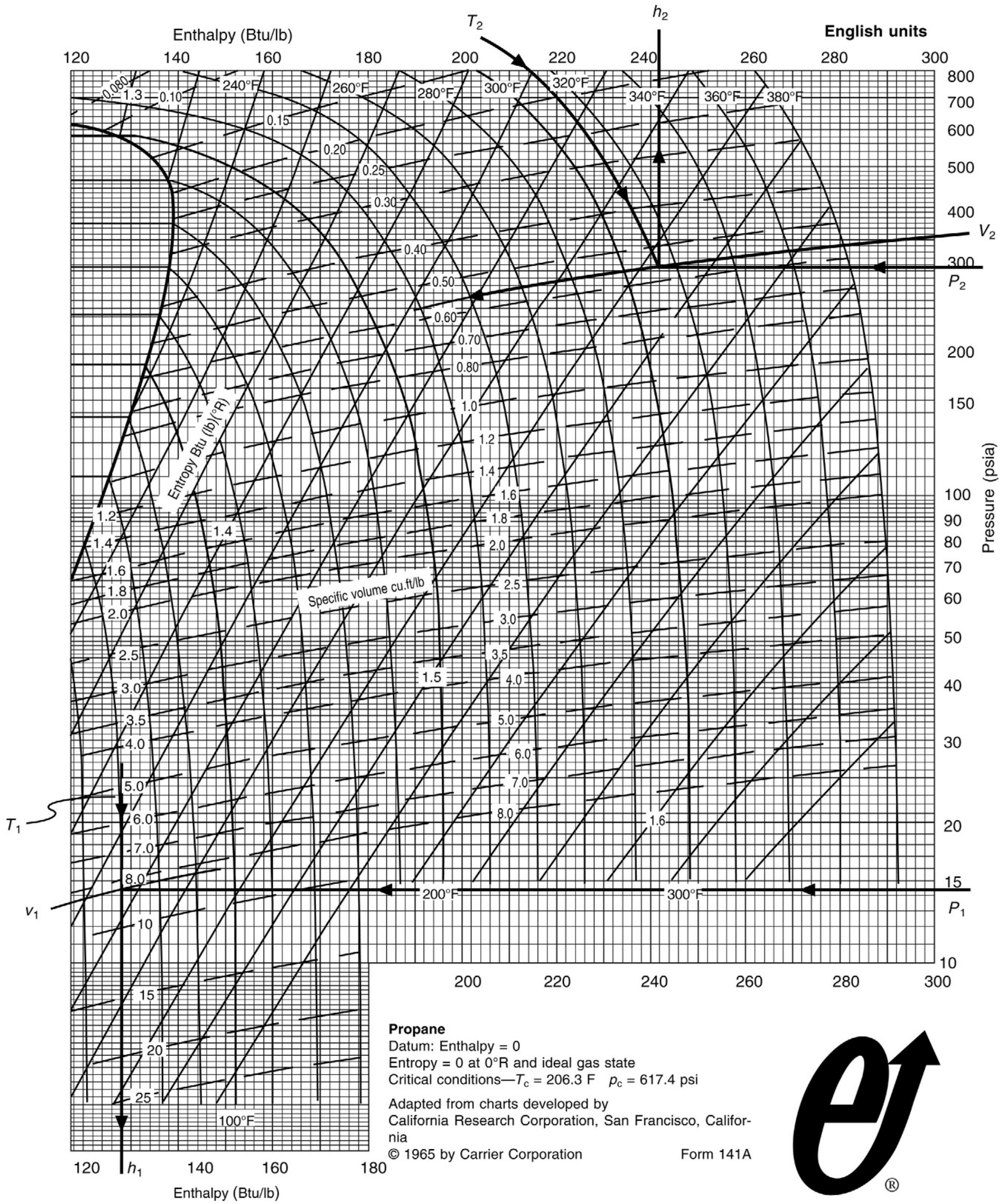


FIG. 7.11 Mollier diagram for propane (see Example 7.3).

Polytropic efficiency

$$\eta_p = \frac{H_P}{(h_2 - h_1)\,778.16} = \frac{64{,}800}{(242-128)778.16} = 0.73 \quad \text{(Ref. 0.726)} \tag{7.6}$$

Gas horsepower

$$\dot{M} = \frac{Q}{v} = \frac{3212}{8.4} = 382\#/\text{min}$$

$$\text{GHP} = \frac{H_P \dot{M}}{\eta_p 33{,}000} = \frac{64{,}800 \times 382}{0.73 \times 33{,}000} = 1027\,\text{HP} \tag{7.12}$$

Power balance

Motor power

$$\text{HP} = E \times I \times \eta \times \text{P.F.} \times 1.732/746$$
$$= 6600 \times 85 \times 0.95 \times 0.91 \times 1.732/746$$
$$= 1125$$

Mechanical losses

45 compressor bearings and seals
12 motor bearings
22 gear
______
79 total

$$\text{Test Error} = \left[\left(\frac{\text{Driver Power} - \text{Losses}}{\text{GHP}}\right) - 1\right] \times 100\%$$
$$= \left[\left(\frac{1125-79}{1027}\right) - 1\right] \times 100\%$$
$$= 1.9\%$$

Example 7.4

Hand Calculate Gas Properties [11]

Gas Analysis

Propane	0.89
Butane	0.06
Ethane	0.05

$P_1 = 20$ psia	$T_1 = 40°\text{F} = 500°\text{R}$
$P_2 = 100$ psia	$T_2 = 180.5°\text{F} = 640.5°\text{R}$
$N = 10650$ RPM	
$Q = 5280$ ICFM	

Use Table 7.1 to calculate

- MW of mixture
- k of mixture
- Z of mixture

TABLE 7.1 Calculation of Properties for a Gas Mixture

Gas Mixture	(1) Mol% each gas	(2) Mols/h (Mol% × 2400)	(3) Mol. Mass	(4) (1) × (3)	(5) Mass% [(4) ÷ 44.23] × 100	(6) T_c (°R)	(7) p_c (psi)	(8) (1) × (6)	(9) (1) × (7)	(10) $C_{p,m}$ (Btu/mol-F)	(11) (1) × (10)
Propane	89%	2136	44.09	39.24	88.72%	666	617	592.7	549.1	16.58	14.76
n-Butane	6%	144	58.12	3.49	7.89%	766	551	46.0	33.1	22.53	1.35
Ethane	5%	120	30.07	1.50	3.39%	550	708	27.5	35.4	11.98	0.60
		2400		44.23				666.2	617.6		16.71
$k\,(\text{Mixture}) = \frac{16.71}{16.71 - 1.99} = 1.135$				Apparent Mol. mass of mixture				T_c (mix)	p_c (mix)		$C_{p,m}$ (mix)

Note: Items 3, 6, 7, and 10 are obtained from Table A.1 in Appendix A.

Find Z_1 using Fig. A.1. First find P_{R_1} and T_{R_1}:

$$P_{R_1} = P_1/P_c \, T_{R_1} = T_1/T_c$$

$$P_{R_1} = \frac{20}{617} = 0.0324 \quad T_{R_1} = \frac{40+460}{666} = 0.75$$

$Z_1 = 0.97$ (from Appendix A, Fig. A.1) (Ref. 0.970)

$$v_1 = ZRT/144P \tag{7.19}$$

$$v_1 = 0.97 \times \frac{1545}{44.23} \times \frac{(40+460)}{144 \times 20}$$

$$= 5.88\,\text{ft}^3/\text{lb} \quad (\text{Ref. } 5.884)$$

$$P_{R_2} = \frac{P_2}{P_c} = \frac{100}{617.0} = 0.162$$

$$T_{R_2} = \frac{T_2}{T_c} = \frac{640.5}{666.0} = 0.961$$

From Fig. A.1, $Z_2 = 0.93$ (Ref. 0.9323)

$$v_2 = 0.93 \times \frac{1545}{44.23} \times \frac{640.5}{144 \times 100}$$

$$= 1.44\,\text{ft}^3/\text{lb} \quad (\text{Ref. } 1.449)$$

Head

$$H_P = 72\left[\ln\left(\frac{P_2}{P_1}\right)\right](P_1 v_1 + P_2 v_2)$$

$$= 72\left[\ln\left(\frac{100}{20}\right)\right](20 \times 5.88 + 100 \times 1.44) \tag{2.36}$$

$$= 30300 \quad (\text{Ref. } 30578)$$

Efficiency

$$\eta_p = \left(\frac{k-1}{k}\right) \div \left(\frac{n-1}{n}\right) \tag{7.7}$$

where

$$n = \frac{\ln(P_2/P_1)}{\ln(v_1/v_2)} = \frac{\ln(100/20)}{\ln(5.88/1.44)} = 1.144$$

$$\eta_p = \left(\frac{1.135-1}{1.135}\right) \div \left(\frac{1.144-1}{1.144}\right) = 0.95 \quad (\text{Ref. } 0.716)$$

Obviously, this is an incorrect answer. This compressor cannot possibly have an efficiency of 95%! So we should go back and recalculate k using an average temperature.

$$T_{\text{average}} = \frac{T_1 + T_2}{2}$$
$$= \frac{40 + 180.5}{2}$$
$$= 110^{\circ}\text{F}$$

From Table A.1 for propane:
Cp, m[6] = 16.82 @ 50°F, 23.57 @ 300°F
Interpolate for value @ 110°F

$$\frac{110-50}{300-50} = \frac{X-16.8}{23.57-16.8}$$
$$\frac{60}{250} = \frac{X-16.8}{6.77}$$
$$X = 1.63 + 16.8$$

Cp, m @ 110°F = *X* = 18.4
Values can be obtained for butane and ethane in a similar fashion.

Butane
Cp, m = 24.81 @ 110°F
Ethane
Cp, m = 13.14 @ 110°F

To find k at the average temperature, first find *Cp, m* (mix) using the *Cp, m* values at average temperature just calculated.

$$\begin{aligned} Cp,\, m\,(\text{mix}) &= 0.89 \times 18.4 + 0.06 \times 24.81 + 0.05 \times 13.14 \\ &= 16.38 + 1.49 + 0.67 \\ &= 18.52 \end{aligned}$$

$$\text{mixture} = \frac{18.52}{18.52 - 1.99} \tag{2.54}$$
$$= 1.12$$

Using this new value of k for average conditions calculate Eff.

$$\eta_p = \frac{(k-1)/k}{(n-1)/n}$$
$$= \frac{(1.12-1)/1.12}{(1.144-1)/1.144} \tag{7.7}$$
$$= 0.085 \quad (\text{Ref. } 0.716)$$

This is still an inaccurate evaluation of the compressor efficiency. (Note the reference value of .716 evaluated by Gas Flex, a gas properties program. Gas Flex uses the Mallen-Saville equation (7.4d). The k value is not involved in this calculation, thus there is no issue with the value of k not being constant.)

The efficiency cannot accurately be hand-calculated for this problem. This is common for high-mole-weight gases. The problem is due to the nonlinear relationship of the gas properties near the dew point. When looking at values far from the dew point, such as with air or nitrogen, the values are near linear and perfect gas laws are accurate. See the Mollier diagrams in the reference section and the situation will be apparent.

Power

Since efficiency cannot accurately be established, then also work and power cannot be established. We can, however, work backward from the driver to establish the gas power.

Driver—Steam Turbine (from Fig. 7.12):

t_1 = 670°F	P_1 = 400 psia
t_2 = 411°F	P_2 = 100 psia
h_1 = 1345	h_2 = 1235

Steam flow = 506 #/min

6. *Cp*, *m* and Mc_p are used interchangeably. See "Gas Mixtures" in Chapter 2.

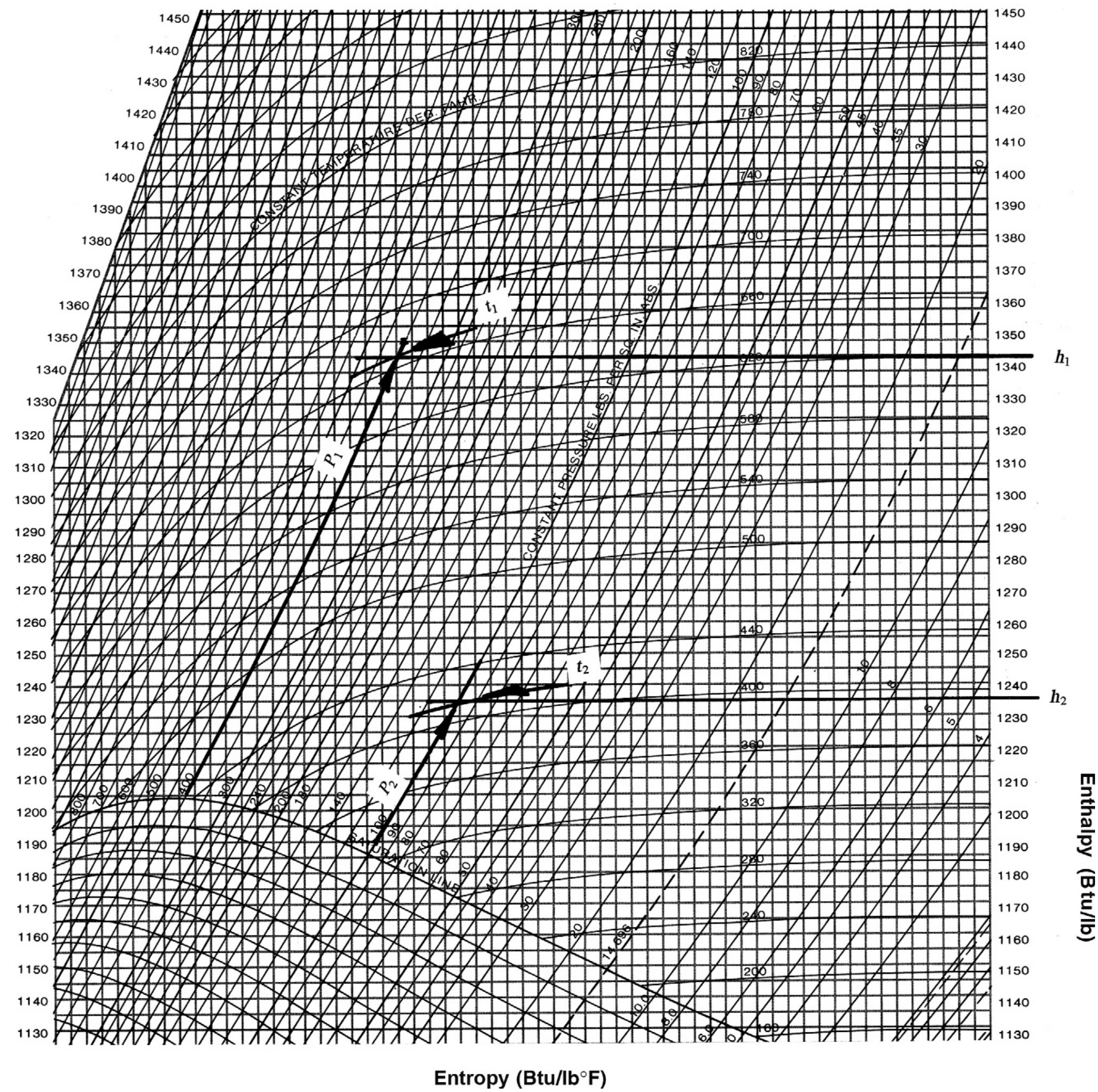


FIG. 7.12 Mollier diagram for steam (Example 7.3).

$$
\begin{aligned}
\text{GHP} &= \dot{M}(h_1 - h_2)/42.4 \\
&= 506(1345 - 1235)/42.4 \qquad (7.11) \\
&= 1313\,\text{HP}
\end{aligned}
$$

Mechanical losses
(Estimated losses)
65 compressor
53 turbine

---

118 total
GHP = 1313 − 118 = 1195 HP
(Compressor gas horsepower)

Compressor Efficiency (see Eqs. 2.45–2.47)

$$\text{Efficiency} = \frac{\text{Head}}{\text{Work}}$$

$$\text{Work} = \frac{\text{GHP}}{\dot{M}}$$

$$\dot{M} = Q/v_1$$

$$= \frac{5280\,\text{ft}^3}{\text{min}} \times \frac{\text{lb}}{5.88\,\text{ft}^3}$$

$$= 898\,\text{lb/min}$$

$$\text{Work} = 1195\,\text{HP} \times \frac{\text{min}}{898\,\text{lb}_\text{m}} \times \frac{33{,}000\,\text{ft-lb}_\text{f}}{\text{HP-min}}$$

$$= 43{,}900\,\frac{\text{ft-lb}_\text{f}}{\text{lb}_\text{m}}$$

$$\eta = \frac{30300}{43900} = 0.69 \qquad (\text{Ref. } 0.716)$$

Now plot data on nondimensionalized curves to minimize effect of small speed changes (see Fig. 7.13). Other types of curves that compensate for speed are shown in Figs. 7.14 and 7.15.

$$H/N^2 = 30{,}300/10{,}650^2 = 2.67 \times 10^{-4}$$
$$Q/N = 5280/10{,}650 = 0.498$$

Plotting the data on the curves shows that both efficiency and head are lower than but close to predicted values and the work input is per prediction; therefore, the test was probably of reasonable accuracy.

Note: An alternate way to measure horsepower is to use a coupling equipped with a torque meter. Some useful equations are:

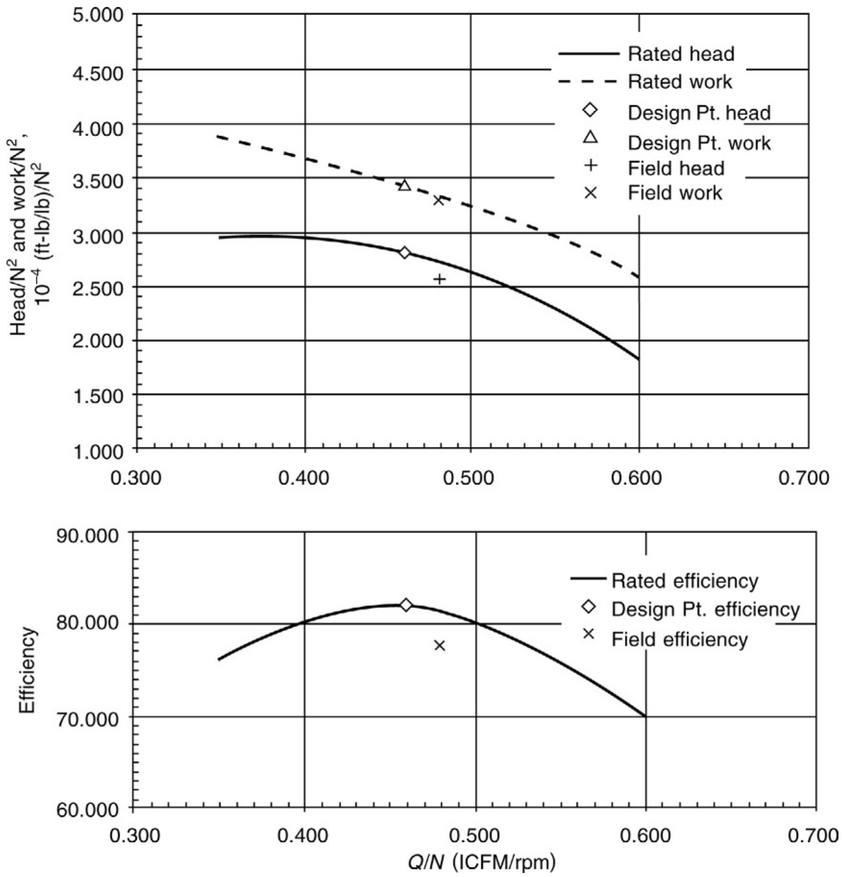


FIG. 7.13 Nondimensionalized performance curve (Example 7.4). While a plot of head coefficient and work coefficient (see page 21) versus flow coefficient is the "proper" nondimensionalized curve, simply dividing the head and work by speed squared and plotting versus flow divided by speed gives the same effect. The idea is to minimize the effects of speed differences during testing. Another preferred method is to simply adjust the values using the fan laws and noting on the plot that this has been done.

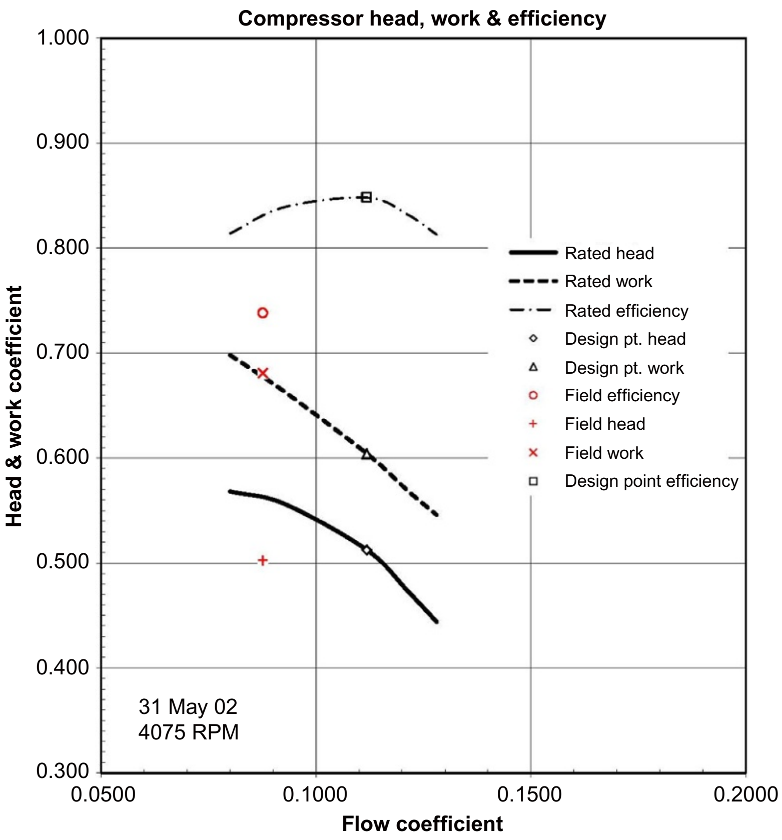


FIG. 7.14 Typical coefficient curve. Plot data in coefficient form (see Performance Coefficients, Chapter 2) and speed effects are limited. See also Figs. 4.16 and 4.18.

$$\begin{aligned} \text{HP} &= 550\,\text{ft-lb/s} = 33{,}000\,\text{ft-lb/min} \\ &= \frac{\text{torque} \times \text{speed} \times 2\pi}{33{,}000} \\ &= \text{torque} \times \text{speed}/5252 \end{aligned}$$

where

$$\begin{aligned} \text{Torque} &= \text{ft-lb} \\ \text{Speed} &= \text{rpm} \end{aligned}$$

From the torque meter reading between the turbine and compressor:

$$\begin{aligned} \text{Torque} &= 605\,\text{ft-lb} \\ \text{SHP} &= \frac{605 \times 10{,}650}{5252} \\ &= 1230\,\text{HP} \end{aligned}$$

Power balance

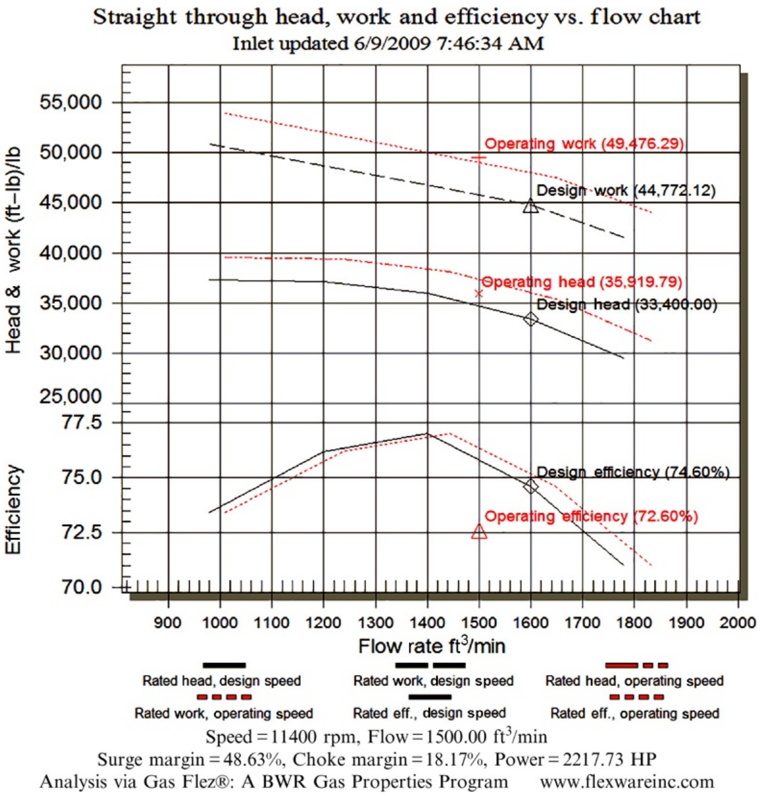
From Torque meter :

$$\begin{aligned} \text{SHP} &= 1230 \\ \text{GHP} &= 1230 - 65 \\ &= 1165\,\text{HP} \end{aligned}$$

From steam tables :

$$\begin{aligned} \text{GHP} &= 1195 \\ \text{Test error} &= \left(\frac{1195}{1165} - 1\right) 100\% \\ &= 2.6\% \end{aligned}$$

FIG. 7.15 Typical curve for plotting single point compressor performance. Two speed lines are shown. The original design speed curves, plus a new set of curves adjusted to the current operating speed. The performance curves have been corrected for speed using the fan laws and are continuously adjusted for the current operating speed as the speed changes.



Chapter 8

Multisection Compressors

With the proper procedures and tools, field testing multisection compressors is not a difficult task. While Chapter 7 will give you the tools and guidance you need to conduct a proper field test, there are additional complications relating to multi-section compressors that need to be considered. This chapter will give you the additional precautions you will need to reduce and properly assess multisection compressor test data.

ISOCOOLED COMPRESSORS

Field testing isocooled compressors can be straightforward, but a few precautions are in order. Treat each section of the machine like a separate single-section compressor while keeping the following items in mind.

Gas Analysis

The composition of the process gas flowing to the second section of the compressor may be different from the process gas in the first section. Liquids may form in the cooler and be drained out prior to continuing on to the next section. Flow to the second section thus will have a lower mass flow rate and the gas will have a different mole weight. This is especially true for installations like wet (rich) gas compressors. Sideload flows or intermediate processes may also change the composition of the process gas.

Measuring the flow of liquid flowing from the knockout drum and the process gas flow at the compressor main inlet (or discharge) will provide the mass flow rate for the other section by subtracting (or adding). For example, in a three-section wet-gas compressor (shown in Fig. 8.1), sample the liquid knockout from each drum to calculate the mass balance around each compressor section. The measured gas compositions should meet the following equation:

$$m_1 = m_2 + m_3 = m_3 + m_5 + m_6 = m_3 + m_6 + m_8 + m_9 \tag{8.1}$$

The mass flow rate of noncondensables such as hydrogen and methane should be constant from section to section.

Be sure to follow proper precautions when taking a gas sample. Condensate can form on the gas sample container walls and give erroneous results unless proper procedures are followed. The gas bomb should be heated to the temperature of the gas being sampled during the sampling process and then reheated to that same temperature before transferring the gas to the gas chromatograph to assure condensation is not a factor (see gas sampling, Chapter 7). Make sure you confirm values by comparing the discharge gas analysis to the inlet gas analysis for the same section. The accuracy of your test results is no better than the agreement between your gas analysis results.

Example 8.1

Isocooled Wet-Gas Compressor

Two sets of test data were taken. Gas samples were collected from the compressor suction and discharge nozzles before and after the test. Liquid samples and flow rates from the isocooler condensate were obtained as well. Computer-predicted liquid knockout agreed with second section analysis. The gas analysis was modified assuming the gas to be saturated with water at the suction.

Test results (Figs. 8.2 and 8.3 and Table 8.1):

- There was a 9% scatter in results (head) due to a disparity in the gas analysis at the inlet versus the gas analysis at the discharge nozzle (see Figs. 8.2 and 8.3).
- For the lighter gas analysis, each section was 9%–10% low in head.
- Based on calculated compressor power, the turbine efficiency is 62.6% (vs. 75% design efficiency).

Compressor Performance. https://doi.org/10.1016/B978-0-12-814219-6.00008-8

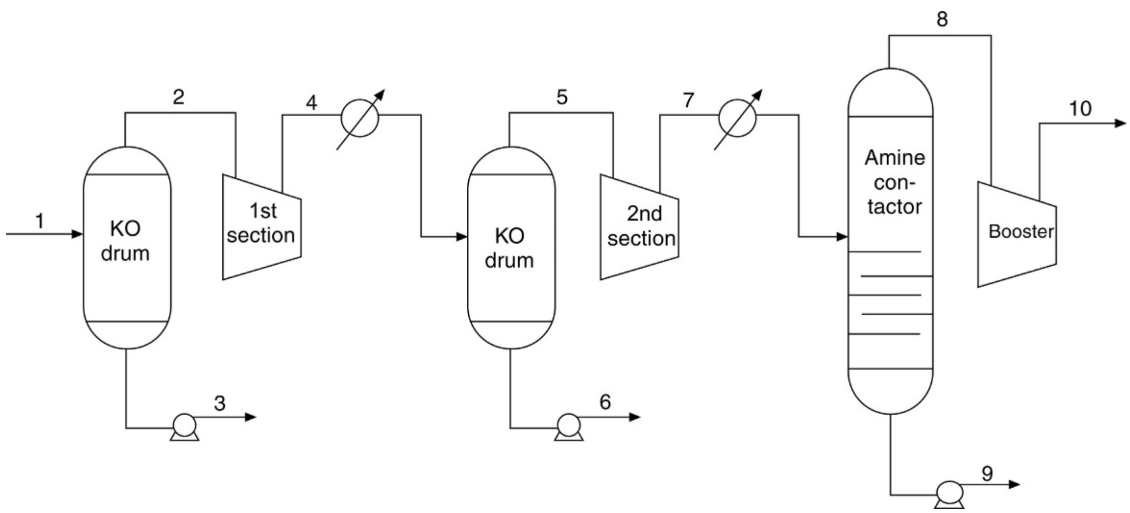


FIG. 8.1 The flow diagram for a three-section wet-gas compressor. Sample the liquid knockouts to calculate the balances around every process split. The measured gas compositions should meet the constraints of Eq. (8.1) [33].

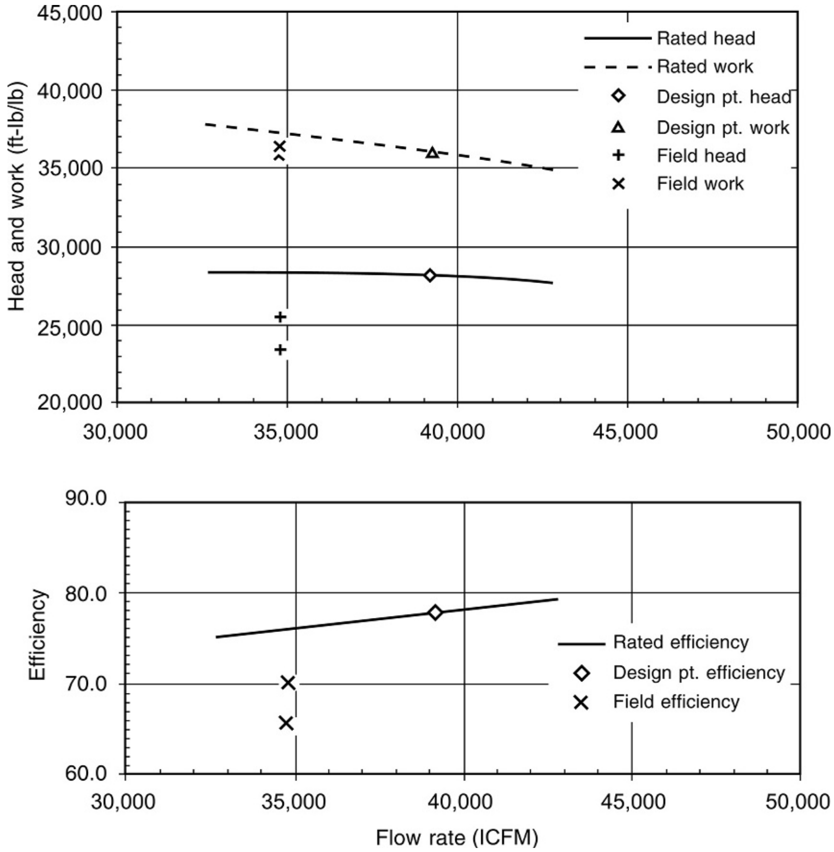


FIG. 8.2 Section 1 of a wet-gas compressor, Example 8.1. Difference in data points is due to wide variation in gas sample data. Data have been fan law corrected to design speed conditions.

Discussion

It is rather obvious that the gas analysis was not accurate (the 9% data scatter). In spite of the large errors in this test, some conclusions can be drawn from the results. It should be noted that discarding the higher mole weight gas samples and using only the lower mole weight gave better results (closer to predicted compressor performance). Using the lower mole weights, the head is 9%–10% low in each section and the work input is within 4%. While the compressor performance was found to be off spec, the turbine was found to be underperforming also (efficiency: 62.6%). Most likely, this low efficiency is caused by fouling or erosion of the turbine blades.

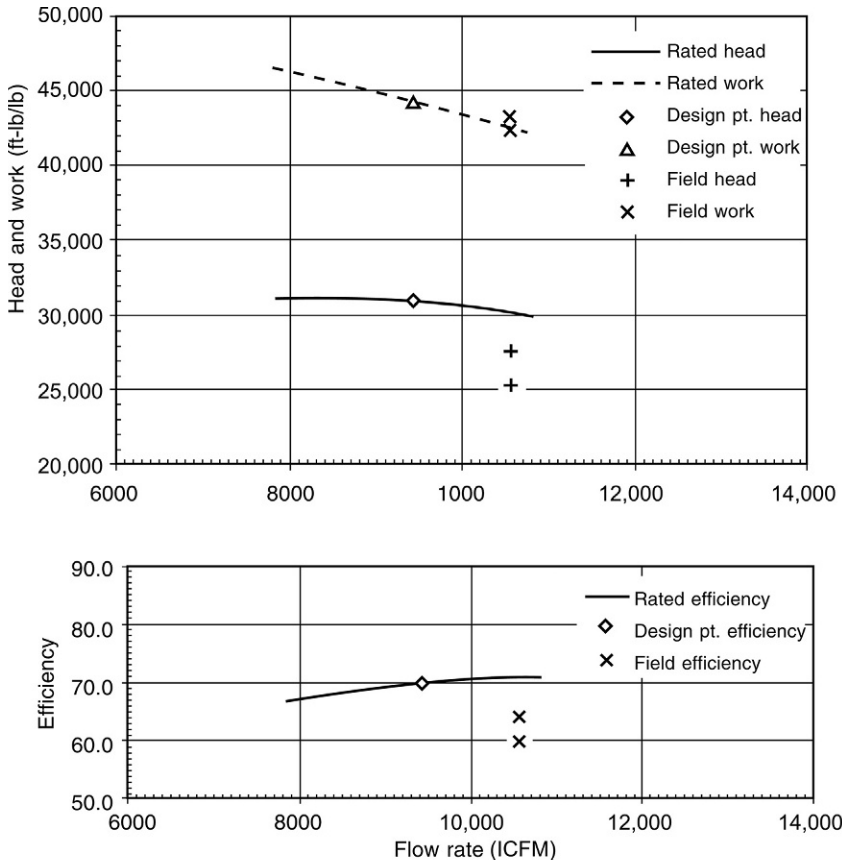


FIG. 8.3 Section 2 of a wet-gas compressor for Example 8.1. Data have been fan law corrected to design speed conditions.

The main problem with this test was the gas analysis. There was concern that the test bombs may have leaked out some “light ends,” making the gas heavier than actual. The gas was not immediately analyzed, thus making this feasible. This would explain why the lighter analysis matched prediction better. Another possibility is that the test bombs were not heated to gas temperatures before receiving the gas into the bombs. This could have caused condensation on the bomb walls, thus making the sample heavier than the actual process gas. Another concern is the water content in the gas. Note that the gas sample was not tested for water content and it was assumed that it was saturated with water at the compressor inlet conditions.

Considerations for improvements to this test

- Test for water in the gas.
- Test gas bombs for leakage before use.
- Conduct gas analysis immediately after taking the sample.
- Heat test bomb to process gas temperature prior to receiving gas into the bomb.

Heat Transfer

Heat conduction from the discharge of the first section to the next section inlet can make results confusing. For example, assume that a compressor has a discharge temperature for the first section of 248°F and an inlet to the second section of 55°F (Fig. 8.4). It is easy to understand that there is considerable heat flowing across the intermediate wall separating the two sections because of this high temperature differential. This heat is flowing from the discharge of the first section to the inlet of the second, lowering the discharge temperature of the first section and raising the inlet temperature (to the first impeller) of the second section. The measured temperature at the discharge flange of the first section thus does not accurately represent the true temperature at the discharge of the last wheel of that section. Likewise, the temperature at the inlet flange to the second section does not represent the true temperature at the first impeller in the second section due to the heat transfer effect.

TABLE 8.1 Data for Steam Turbine-Driven Wet-Gas Compressor

	Compressor Data			
	Section 1		Section 2	
	pt1	pt2	pt1	pt2
Pressure, psig				
Inlet	10.1	10.1	54.5	54.5
Discharge	60	60	207.5	207.5
Temperature, °F				
Inlet	107	107	104	104
Discharge	221	221	241	241
Flow, icfm	35,600	35,600	10,804	10,804
Head, ft-lb/lb	24,610	26,767	28,948	26,478
Efficiency, %	65.6	70.3	64.2	59.7
Work input ft-lb/lb	37,515	38,075	45,090	44,352
MW	42.4	39.2	35.8	38.8

	Turbine Data	
	Test Data	Rated Conditions
Inlet pressure, psig	585	610
Inlet temperature, °F	737	700
Exhaust pressure, Hg	4	4
Speed, rpm	4565	4460
Steam flow rate, #/h	128,000	94,300
Turbine power, HP	13,685[a]	12,306[a]
Compressor power, HP	13,685[a]	12,306[a]
Turbine efficiency, %	62.6	75.0

[a]*Note mechanical losses were not considered.*

The first section discharge temperature is artificially low; thus, a higher-than-actual efficiency and corresponding low power is calculated. Just the opposite is true for the second section. While this effect should be reflected in the manufacturer's predicted performance values, the actual heat transfer may vary.

Example 8.2

Isocooled Chlorine Compressor
High discharge temperatures prompted an analysis of this compressor (Fig. 8.4).

Discussion of Results

The overall head and gas power (Table 8.2) were calculated by adding the sectional values. The overall efficiency was calculated using the overall head and work input.

While the discharge pressures were close to predicted values, the discharge temperature for the first section is consistently lower than predicted values while the discharge temperature for the second section is significantly higher than predicted values. As a result, the first section efficiency is high, while the second section is low in efficiency (Figs. 8.5 and 8.6).

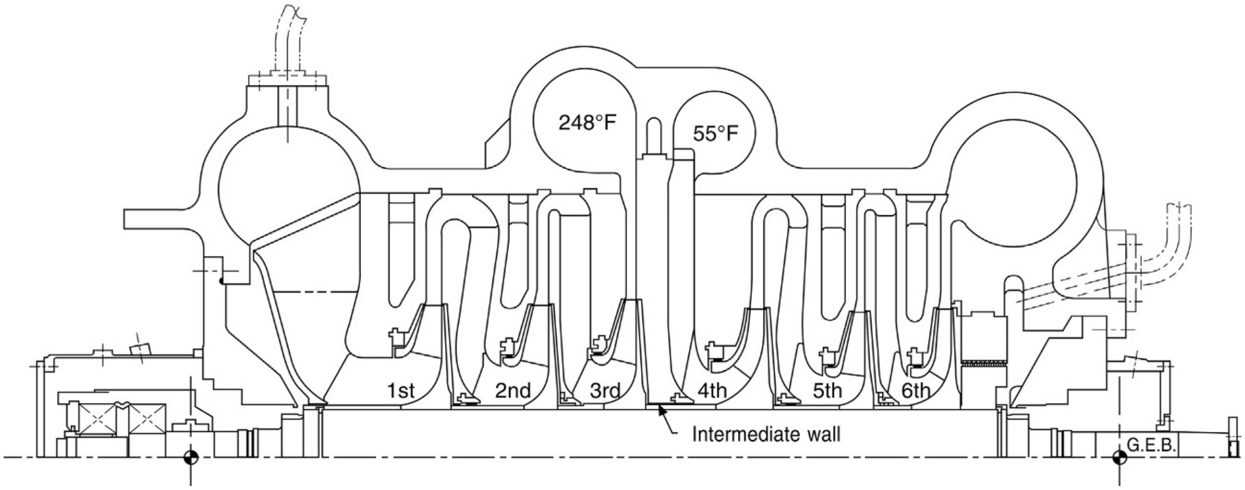


FIG. 8.4 Isocooled chlorine compressor (Example 8.2). The discharge temperature of the first section is 248°F and the inlet to the second section is 55°F. The noninsulated wall between the two sections can transfer significant heat resulting in misleading data.

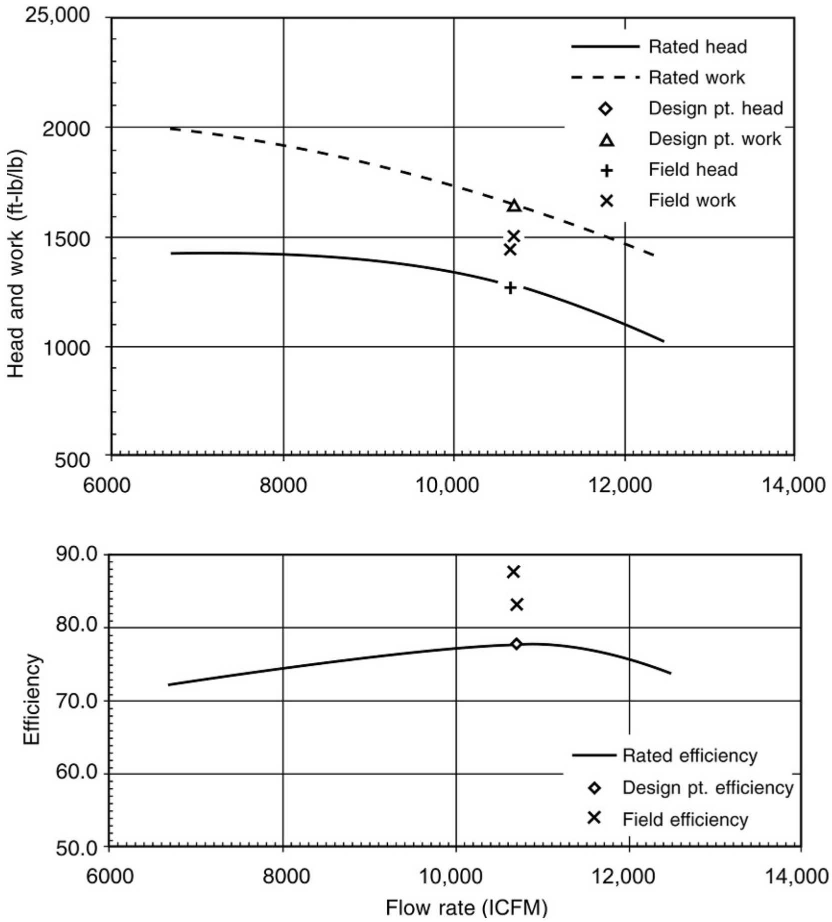
TABLE 8.2 Data for Motor-driven Isocooled Chlorine Compressor

	Compressor Data			
	Section 1		Section 2	
	pt1	pt2	pt1	pt2
Pressure, psia				
Inlet	11.8	12.5	25.8	27.7
Discharge	30.2	30.7	71.7	71.7
	30.7[a]	30.4[a]	73.6[a]	75.3[a]
Temperature, °F				
Inlet Discharge	66.8 232.7 248.4[a]	74.8 226.1 243.3[a]	54.2 264.7 253.4[a]	55.6 259.3 244.5[a]
Flow, icfm	10,710	10,360	4916	4632
Head, ft-lb/lb	12,438 12,788[a]	11,912 11,918[a]	13,515 13,737[a]	12,526 13,017[a]
Efficiency, %	83.2 77.8[a]	87.5 78.2[a]	71.4 77.2[a]	68.3 77.3[a]
Work input ft-lb/lb	14,950 16,437[a]	13,614 15,240[a]	18,928 17,794[a]	18,340 16,480[a]

	Overall Results	
	pt1	pt2
Head, ft-lb/lb	25,953 26,525[a]	24,438 24,935[a]
Efficiency, %	76.6 77.5[a]	76.5 77.8[a]
Work input, ft-lb/lb	33,877 34,211[a]	31,953 32,068[a]
Gas power, HP	1667 1684[a]	1558 1596[a]
Speed MW = 70.7	5351	5189

[a]*Predicted values.*

FIG. 8.5 Performance curve for the first section of the chlorine compressor in Example 8.2. Note that the head is near prediction but the work input is very low and the efficiency is high.



The overall results (Table 8.2), however, are very close to predicted values. These results may indicate a higher-than-predicted heat transfer between the intermediate diaphragm separating the two sections of the compressor. It is unlikely that fouling is a factor as the overall values are near predicted levels.

Seal Leakage

To fully understand the performance of isocooled compressors, seal leakage must also be considered. The two areas of importance are the balance piston seal and the seal between the isocooled sections. Normal seal leakage is represented in the compressor design performance. Seal degradation will affect the observed power and efficiency values. Higher than design flow across the balance piston seal will affect the discharge temperature of the *first* section since the hot balance piston leakage will heat up the compressor first stage inlet gas (assuming the balance line is returned to the main inlet). The compressor will also be operating further out on the curve due to the high balance piston flow reducing the head. Performance analysis of a compressor with a defective balance piston will thus show a higher-than-normal power consumption (low efficiency) for the first section (see Table 8.3 section one and Fig. 8.8). Also refer to "Balance Piston Seal," Chapter 3. The following sections will also be affected due to the increased flow from the balance piston and the resulting head loss resulting from operation at the higher flow rates.

If the internal seal between the two sections of an isocooled compressor is damaged, increased flow across this seal will affect test results. Higher leakage from the damaged seal will increase the temperature of the second section inlet (after the measurement point). The hot discharge gas of the first section flows to and mixes with the cool inlet gas of the following stage. See Example 8.3, Table 8.3, section 4 and Fig. 8.11.

Leakage across the intermediate wall in a back-to-back compressor will have less effect on the performance than on an inline isocooled compressor. The discharge of one section is leaking into the discharge of the other section and the temperature differential is small. The back-to-back compressor has a much higher pressure differential though, and this can

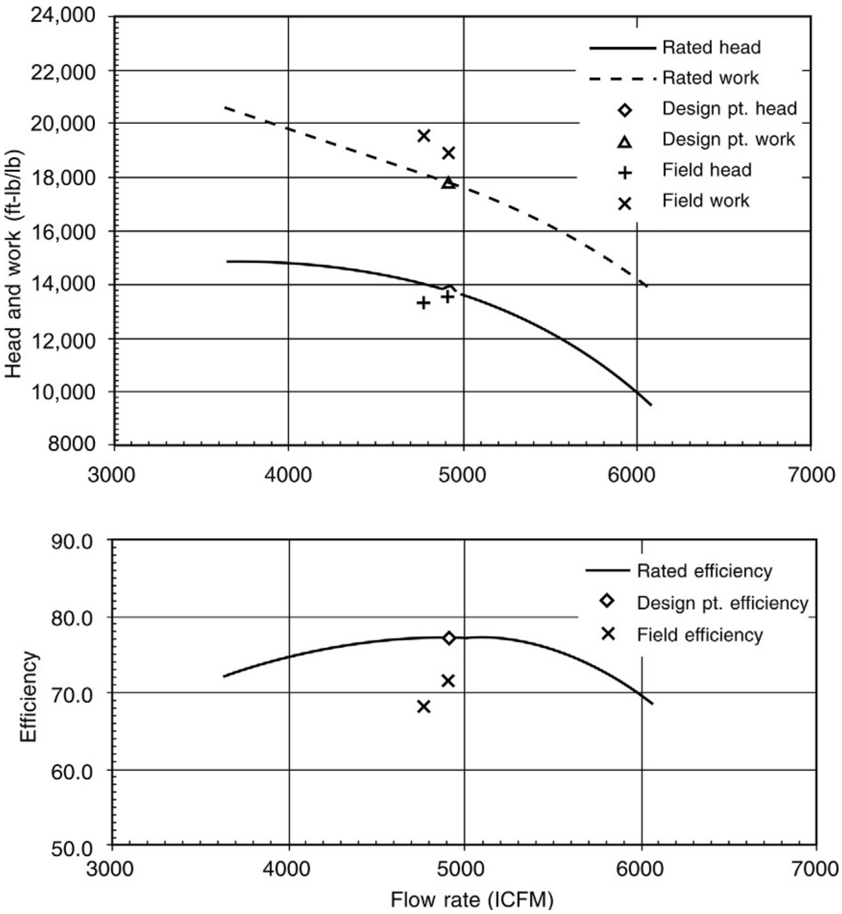


FIG. 8.6 Performance curve for the second section of the chlorine compressor in Example 8.2. Note that the head is near prediction but the work input is high while the efficiency is low, just the opposite of the first section (Fig. 8.5).

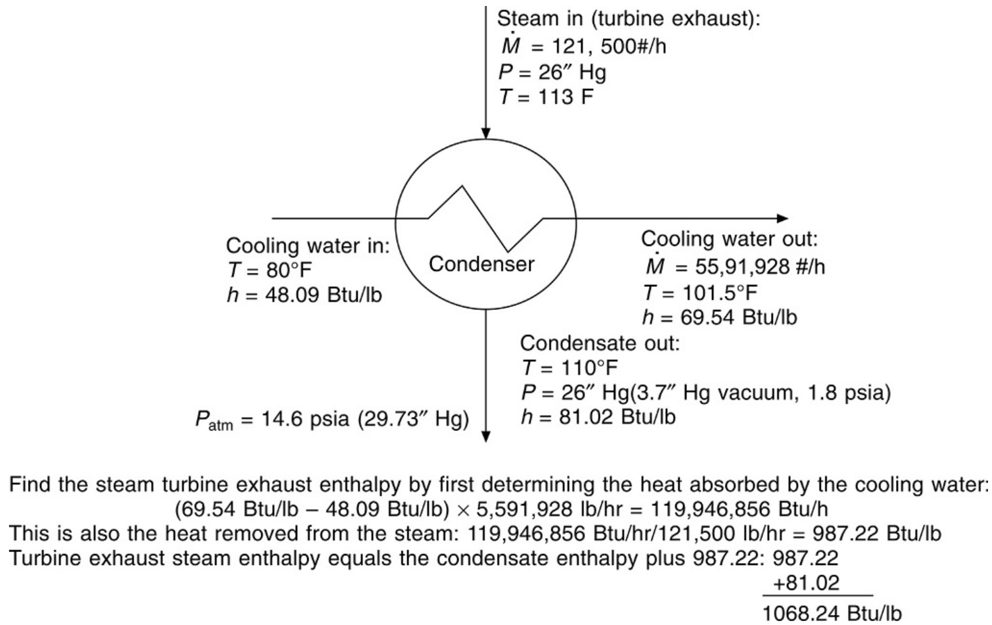


FIG. 8.7 Steam turbine condenser heat balance (Example 8.3). Be especially careful when measuring the cooling water temperature. Very small measurement errors will cause a very large error in the turbine power calculation. Note the temperature rise is very small (21.5°F).

TABLE 8.3 Data for Compressor String Supplying Air to Ammonia Plant (Example 8.3)

Ambient data					
Pressure	psia	14.70			
Temperature	°F	91.00			
Relative humidity	%	95.00			
Flow measurement					
Flow measured by Orifice at section 4 discharge					
Differential pressure	in of H_2O	31.2			
Upstream pressure	psia	525.0			
Upstream temp	°F	357.00			
Pipe ID "D"	in.	14.00			
Beta ratio "d/D"		0.600			
Pressure taps at		Discharge			
Plate material		300 stainless steel			
Inlet Data	**Section**	**1**	**2**	**3**	**4**
Pressure	psia	14.60	42.60	94.60	228.60
Temperature	°F	91.00	88.50	87.00	99.60
Volume flow	SCFM	25,448			
Volume flow	ft^3/min	28,467	9402	4184	1767
Wet mass flow	lb/min	2001.0	1961.2	1950.2	1947.0
Dry mass flow	lb/min	1942.0	1942.0	1942.0	1942.0
Water drop out	lb/min		39.8	11.0	3.2
Relative humidity	%	95.0	100.0	100.0	100.0
Specific humidity		0.0304	0.0099	0.0042	0.0026
Molecular weight		28.45	28.80	28.90	28.93
Discharge Data	**Section**	**1**	**2**	**3**	**4**
Pressure	psia	45.60	96.60	240.60	527.60
Temperature	°F	423.00	294.00	324.00	358.00
Volume flow	ft^3/min	14,609	5701	2358	1119
Section Data	**Section**	**1**	**2**	**3**	**4**
Head	ft-lb/lb	40,193	27,096	31,227	28,189
Efficiency	%	63.08	69.55	69.74	57.80
Gas power	HP	3864	2315	2646	2877
Overall data					
Head	ft-lb/lb	126,705			
Efficiency	%	64.57			
Gas power	HP	11,703			

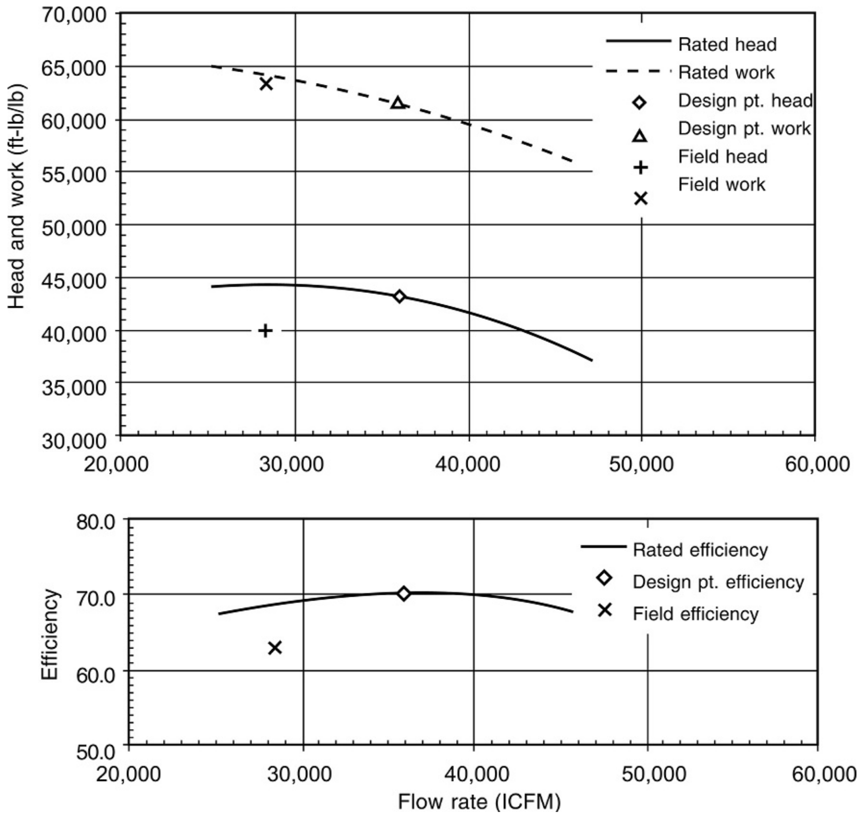


FIG. 8.8 First section of air compressor string, Example 8.3. Note low head and efficiency for this section. A possible cause is the abnormally high balance piston leakage. Data have been fan law corrected to design speed conditions.

Example 8.3

Air Compression String for Ammonia Plant

The operators of an ammonia plant complain that it is necessary to operate the turbine driver of a four-section air compression string flat out and they still cannot make desired air flow rates. The string of equipment is made up of two isocooled centrifugal compressors driven by a steam turbine. Compressor data are shown in Table 10.3.

---

affect the performance of the second section due to the higher flow. The second section will be operating further out on the curve and thus produce lower head.

Analysis

Calculation of the steam turbine efficiency based on the compressor total power resulted in very poor efficiency (58%). In order to confirm this evaluation, a heat balance was completed on the condenser in order to determine the exhaust enthalpy of the turbine (Fig. 8.7). The turbine power based on the condenser heat balance (Table 8.4) was within reasonable agreement with the total compressor string power (Table 8.5).

Note the low head and efficiency on the first section of compression (Fig. 8.8). A possible explanation is high balance piston seal leakage. This was confirmed by noting changes in the axial position and the thrust-bearing temperature. The fourth section is also low in head and efficiency, while sections 2 and 3 are close to expected values (Figs. 8.9–8.11). One explanation for the low efficiency in Section 4 is that the shaft seal in the intermediate wall separating the two compressor sections is damaged and leaking hot gas from discharge of the third section into the inlet of the fourth section (Fig. 8.12).

The resulting power balance of 5.1% (Table 8.5) confirmed compressor power and directed the more serious corrective actions toward the turbine rather than the compressors. While the compressor sections 1 and 4 were off design and needed corrective measures, the test data point to an even more serious problem in the turbine.

TABLE 8.4 Turbine Data for Example 8.3

Inlet Pressure	614.6	psia
Inlet temperature	630.0	°F
Exhaust pressure	3.70	in Hg a
Inlet steam flow	121,500	lb/hr
Speed	6,750	RPM
Efficiency	58.0	%
Inlet specific volume	0.9599	Ft^3/lb
Inlet enthalpy	1307.8	BTU/lb
Inlet entropy	1.5466	BTU/lb-R
Inlet saturation temperature	488.8	°F
Inlet superheat	141.2	°F
Exhaust enthalpy	1068.0	BTU/lb
Exhaust moisture	4.5	%
Exhaust temperature	121.7	°F
Theoretical steam rate	6.11	lb/HP/h
Steam rate	10.77	lb/HP/h
Power	11,280	HP

Based on the condenser heat balance (Fig. 8.7), efficiency of the turbine is very low, indicating maintenance is required.

TABLE 8.5 Power Balance for Air Compression String, Example 8.3

Compressor gas power	
Section 1	3864
Section 2	2315
Section 3	2646
Section 4	2877
Total	11,703
Mechanical (bearing) losses	182.5
Total absorbed power[a]	11,886
Turbine predicted gas power	14,742
(at operating conditions)	
Difference	2857
% of error	24.0%
Calculated turbine power	11,280
(based on condenser heat balance)	
Difference	606
% of error	5.1%

[a] *This should be equal to the turbine gas power.*

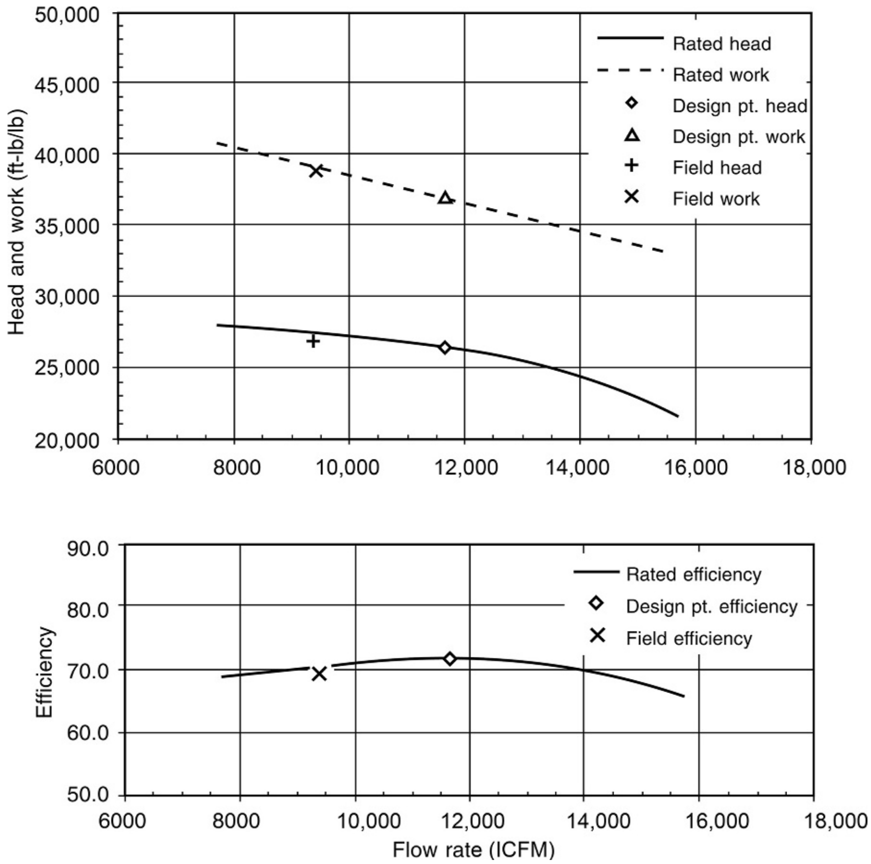


FIG. 8.9 Second section of the air compressor string in Example 8.3. Note all data are close to predicted values. Data have been fan law corrected to design speed conditions.

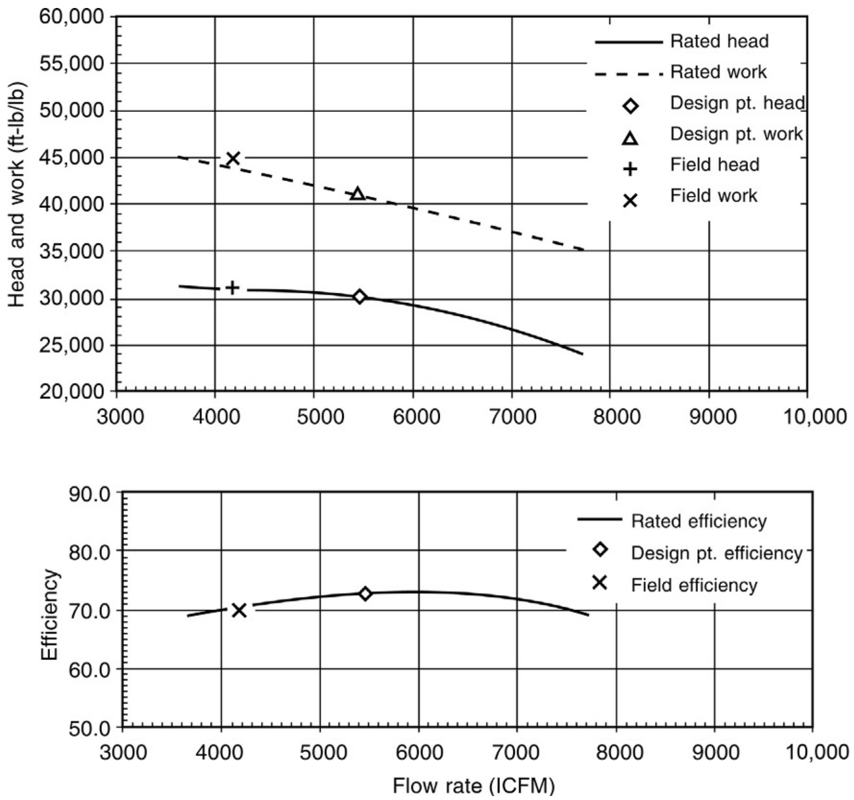
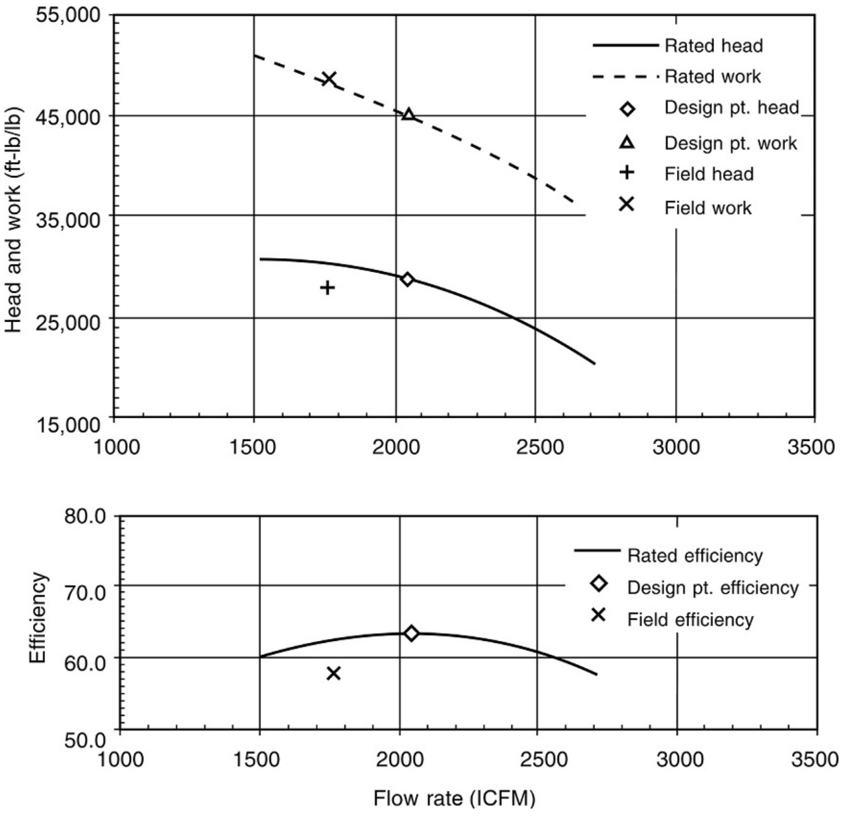


FIG. 8.10 Third section of the air compressor string in Example 8.3. Note all data are close to predicted values. Data have been fan law corrected to design speed conditions.

FIG. 8.11 Fourth section of the air string in Example 8.3. Data have been fan law corrected to design speed conditions. Note that the head and efficiency is low. This may be caused by leakage across the shaft seal of the intermediate wall (Fig. 8.12). Other causes may be buildup or corrosion of the diffuser passages caused by a leaking cooler. Note small diffuser passages, which are very sensitive to corrosion or buildup.



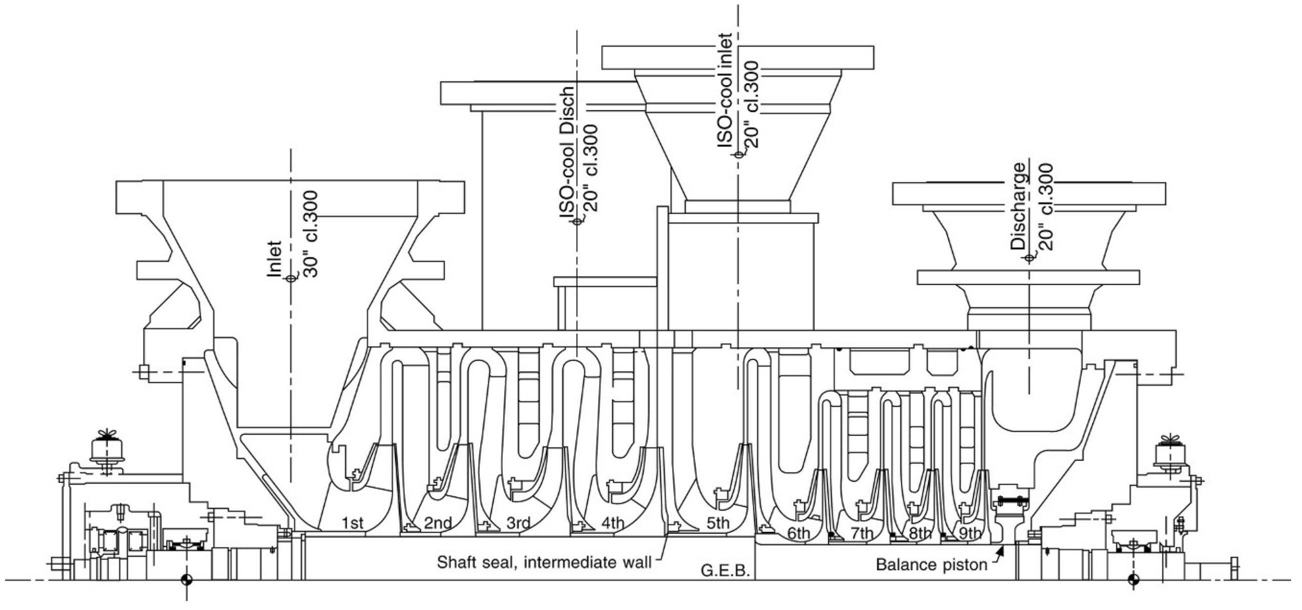


FIG. 8.12 Example 8.3. Cross-section of the second compressor in the air compression string (3rd and 4th sections). Note the small diffuser passages in the last section.

COMPRESSORS WITH ECONOMIZER NOZZLES

Analysis of a sideload compressor requires internal temperature probes in order to properly calculate the performance of the individual sections of the compressor. This is the normal procedure in any shop-proof test following the purchase of a new refrigeration compressor with economizer nozzles since this is the only way that the compressor sectional performance can be properly analyzed and compared to predicted values.

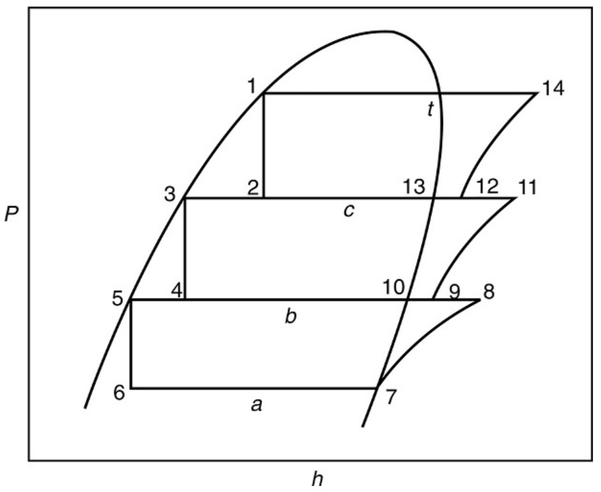


FIG. 8.13 Typical refrigeration heat cycle with economizers.

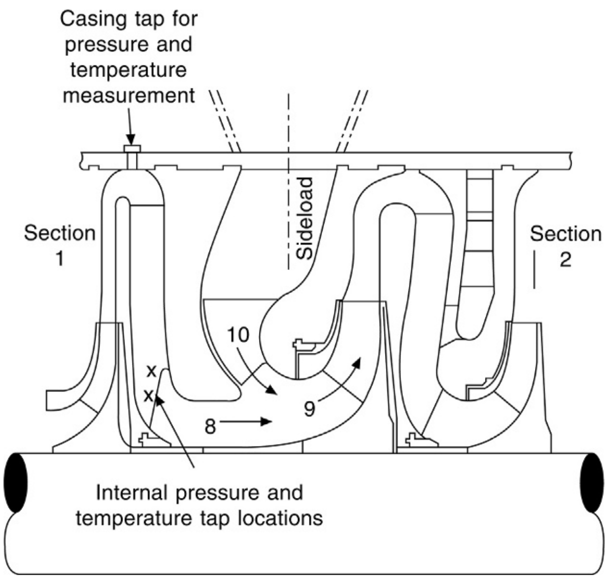


FIG. 8.14 The internal detail of a sideload compressor where the economizer stream meets and mixes with the main refrigeration fluid.

A typical refrigeration cycle with economizers is depicted in Fig. 8.13. As noted, it is necessary to have the measured data for points 7 through 14. Points 8, 9, 11, and 12 are internal areas of the compressor as shown in Fig. 8.14. While pressure can be estimated for the mixing area by estimating the pressure drop from the sideload external measurement point to the mixing area (PTC 10–1997), the internal temperature must be measured.

In a shop test environment, the installed internal probes (Fig. 8.14) are generally removed prior to shipping the equipment and are not available for field measurements. Where these internal data are not available, one option is to use overall values for power and efficiency.

Overall Power

The gas horsepower for a single-stage compressor is:

$$\mathrm{GHP} = \frac{778.16}{33000}(h_2 - h_1)\dot{M} \tag{8.2}$$

where 1 is the inlet conditions and 2 is the discharge.

For a sideload machine:

$$GHP = \frac{778.16}{33000} \Sigma_{1,i} (\dot{M} \Delta h)_i \quad (8.3)$$

where i is the number of sections of the compressor.

For a compressor with two sideloads (three sections):

$$\begin{aligned} \text{GHP} = 0.0236[&\dot{M}_a(h_{14a} - h_7) + \dot{M}_b(h_{14b} - h_{10}) \\ &+ \dot{M}_c(h_{14c} - h_{13})] \end{aligned} \quad (8.4)$$

where (refer also to Figs. 8.13–8.16):
a = Main inlet
b = 1st sidestream
c = 2nd sidestream
t = Total, discharge flow
7 = Main inlet

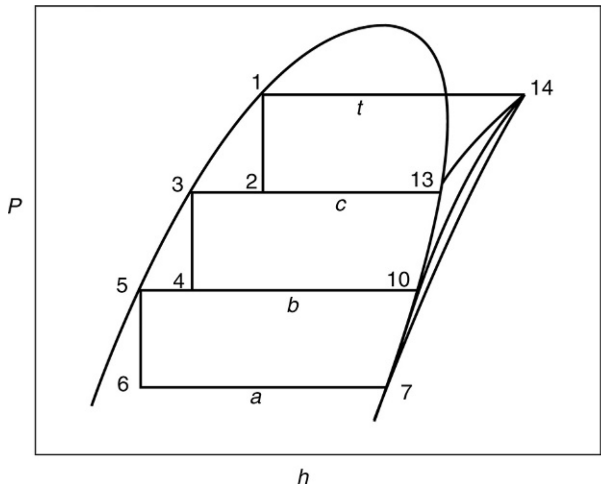


FIG. 8.15 Fig. 8.13 in a simplified "black box" form. The typical economizer cycle is depicted with only external data points.

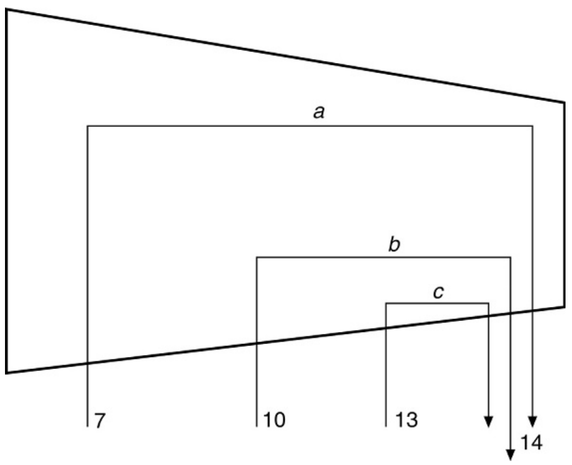


FIG. 8.16 Eq. (8.7) assumes that the sideloads do not mix with the main fluid.

8 = Disch. first section
9 = Inlet second section
10 = 1st sideload
11 = Disch. second section
12 = Inlet third section
13 = 2nd sideload
14 = Final discharge
h = Total enthalpy
H = Head
$\dot{M}$ = Mass flow
W = Work
η = Efficiency, polytropic
GHP = Gas horsepower

Overall Efficiency

Efficiency for a compressor is defined as head divided by the work input:

$$\eta = \frac{H}{W} \tag{8.5}$$

For a sideload refrigeration compressor (Fig. 8.13), an overall compressor efficiency is calculated by dividing the sum of the head from each section by the sum of the work from each section:

$$\eta = \frac{H_{7-8} + H_{9-11} + H_{12-14}}{W_{7-8} + W_{9-11} + W_{12-14}} \qquad 8.6)$$

For the case of the field sideload compressor where points 8 and 11 are not available, a modified cycle is suggested for calculating overall efficiency. A "modified" cycle demonstrated in Figs. 8.15 and 8.16 is offered. Internal mixing is assumed not to occur in order to simplify the solution. The system is treated as three separate and parallel flow paths with measurable end points.

For a single stage compressor:

$$\eta = \frac{H}{W}$$

Multiply by: $\dot{M}/\dot{M}$

$$\eta = \frac{H\dot{M}}{W\dot{M}}$$

For a multisection compressor (three sections):

$$= \frac{H_{7-14a}\dot{M}_a + H_{10-14b}\dot{M}_b + H_{13-14c}\dot{M}_c}{W_{7-14a}\dot{M}_a + W_{10-14b}\dot{M}_b + W_{13-14c}\dot{M}_c}$$

$$W = (\Delta h)778.16 \tag{8.7}$$

$$\eta = \frac{H_{7-14a}\dot{M}_a + H_{10-14b}\dot{M}_b + H_{13-14c}\dot{M}_c}{778.16\left[(h_{14a} - h_7)\dot{M}_a + (h_{14b} - h_{10})\dot{M}_b + (h_{14c} - h_{13})\dot{M}_c\right]}$$

While this procedure may not correctly model the true polytropic process as shown in Fig. 8.13, it does give a very close approximation of the overall condition of the compressor performance.

Any error realized from using Eq. (8.7) in place of Eq. (8.6) is reduced as the main inlet flow is increased and the side stream flows are decreased.

Example 8.4

Using Eq. (8.7). This equation is used when only external data are available:

$$\eta = \frac{H_{7-14a}\dot{M}_a + H_{10-14b}\dot{M}_b + H_{13-14c}\dot{M}_c}{778.16\left[(h_{14a}-h_7)\dot{M}_a + (h_{14b}-h_{10})\dot{M}_b + (h_{14c}-h_{13})\dot{M}_c\right]}$$

$$\dot{M}_a = 3258.5\#/\text{min}\ @ -29.4^{\circ}\text{F}, 18.72\ \text{psia}$$

$$\dot{M}_b = 310.3\#/\text{min}\ @ -6.3^{\circ}\text{F}, 35.03\ \text{psia}$$

$$\dot{M}_c = 1152.7\#/\text{min}\ @ 19.9^{\circ}\text{F}, 58.55\ \text{psia}$$

$$\dot{M}_t = 4721.5\#/\text{min}\ @ 133.8^{\circ}\text{F}, 162.1\ \text{psia}$$

$$H_{7-14} = 35{,}188$$

$$H_{10-14} = 25{,}518$$

$$H_{13-14} = 17{,}126$$

$$h_{14a} - h_7 = 155.4 - 102.8 = 52.6$$

$$h_{14b} - h_{10} = 156.5 - 109.5 = 47$$

$$h_{14c} - h_{13} = 156.7 - 116.5 = 40.2$$

Note that for this example, each section has a different gas analysis. This is why h_{14a} does not equal h_{14b} or h_{14c}.

$$\eta = \frac{35188 \times 3258.5 + 25{,}518 \times 310.3 + 17{,}126 \times 1152.7}{778.16\,[52.6 \times 3258.5 + 47 \times 310.3 + 40.2 \times 1152.7]}$$

$$= \frac{142{,}330{,}030}{180{,}781{,}929}$$

$$= 0.7873$$

In order to check the accuracy of this method, the same data were calculated using Eq. (8.6). This equation is used with internal data and sectional head and work can be accurately calculated:

For Eq. (8.6):

$$\eta = 0.796$$

Error (Eq. 8.6 vs. Eq. 8.7).

$$\text{Error} = \frac{79.6 - 78.73}{79.6} \times 100\%$$
$$= 1.1\%$$

SECTIONAL PERFORMANCE OF SIDELOAD COMPRESSORS

Using the method of matching field work input to predicted values as detailed in Chapter 7, "Special Data Reduction for Sideloads" can give good results for sideload compressors without internal data. The main thing to remember here is that the field work input will probably never exactly match the predicted values. The best that can be done is to assure that each section varies from the predicted work input approximately the same amount.

Example 8.5

Data for a propylene refrigeration compressor are shown in Table 8.6. Begin data analysis by using first the predicted sectional efficiencies. Gas Flex®, a BWR gas properties program, was used for this example. Adjust these sectional efficiency values for each section until the final discharge temperature of the calculation matches the test data discharge temperature. Then plot the sectional data on individual head, work and efficiency versus inlet flow curves.

Further adjustments in the sectional efficiency may be necessary to assure that each section work input varies by the same amount from the predicted value (Figs. 8.17–8.19). This particular compressor shows low head and efficiency in the second section.

TABLE 8.6 Data for Sidestream Refrigeration Compressor

Flange Location		Inlet	SS-1	SS-2	Disch
Inlet flange data					
Pressure	Bar a	1.160	2.590	4.680	
Temperature	°C	–33.2	–19.4	1.9	
Volume flow	m^3/h	24,969	5323	6533	
Mass flow	kg/h	63,601.6	29,649.5	62,588.7	
Compressibility		0.9630	0.9298	0.9012	
Discharge flange data					
Pressure	Bar a				12.780
Temperature	°C				71.8
Volume flow	m^3/h				7223
Mass flow	kg/h				155,839.8
Compressibility					0.8714
Inlet section data					
Compressibility temperature	°C	0.9630 –33.2	0.9465 0.6	0.9214 19.8	
Volume flow	m^3/h	24,969	18,383	17,713	
Mass flow	kg/h	63,601.6	93,251.1	155,839.8	
Discharge section data					
Compressibility		0.9523	0.9318	0.8714	
Temperature	°C	9.7	31.6	71.8	
Volume flow	m^3/h	13,036	11,151	7,223	
Section polytropic data					
Head	N-m/kg	39,593	31,674	56,664	
Efficiency	%	69.00	74.00	81.00	
Gas power	kW	1014	1109	3028	
Total polytropic data					
Head	N-m/kg	127,930			
Efficiency	%	75.2			
Gas power	kW	5151			

Be sure to use BWR equations of state for refrigeration compressors. Otherwise, data reduction accuracy may be compromised.

It may not always be possible to closely match the work input to the predicted values. Excessively high work input can occur when liquids are ingested into the compressor (due to the high fluid density of the gas-liquid mixture). Liquid entrained in the compressor inlet gas is not an uncommon occurrence with refrigeration compressors as liquid is an inherent part of the refrigeration system. Liquid ingestion is especially a concern in rerate and debottlenecking projects where increased velocities through piping and knockout drums result. Besides the potential for mechanical damage, liquid in the gas eliminates the possibility of an accurate performance test. Evaporation of the gas during the compression process reduces the gas temperature via evaporative cooling. The resulting evaporated gas is added to the existing volume of gas in the compressor resulting in volume mismatching (Fig. 4.18), choking the last stages.

FIG. 8.17 Section 1 of a sidestream refrigeration compressor, Example 8.5. Data have been fan law corrected to design speed conditions.

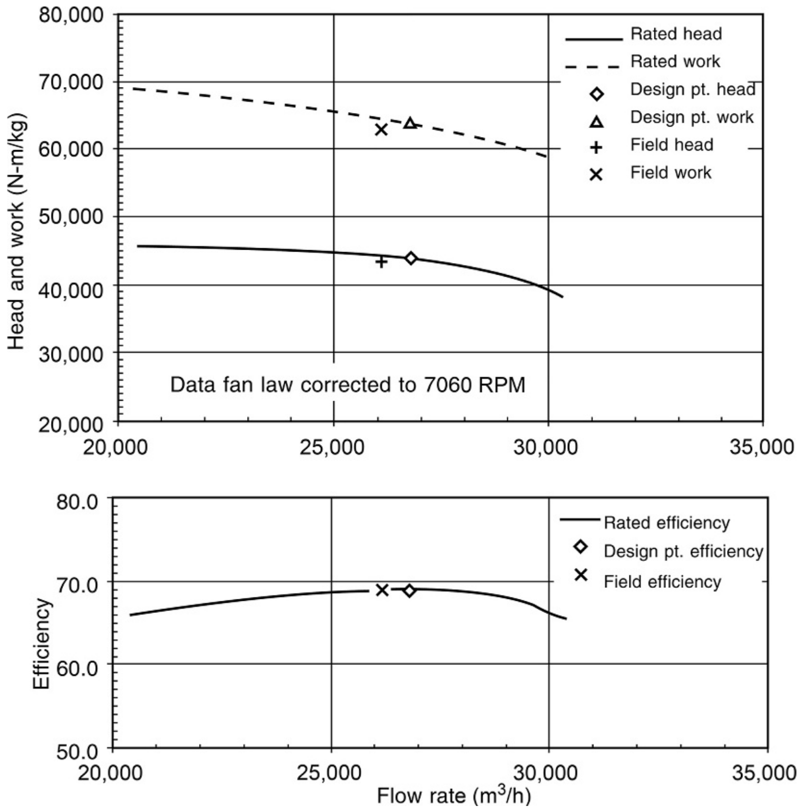
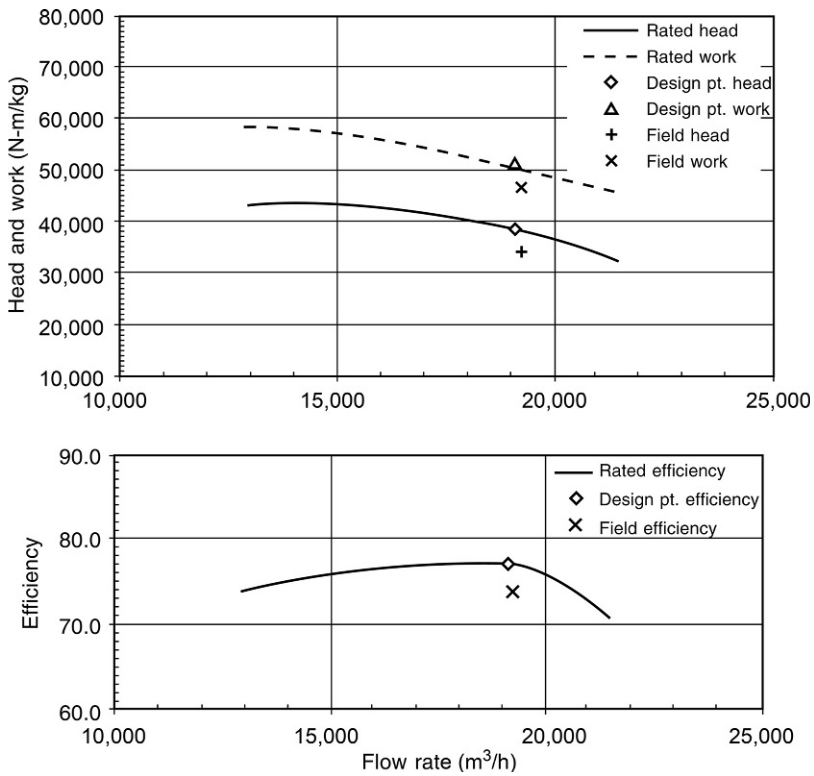


FIG. 8.18 Section 2 of a sidestream refrigeration compressor, Example 8.5. Data have been fan law corrected to design speed conditions. Note head and efficiency are lower than predicted.



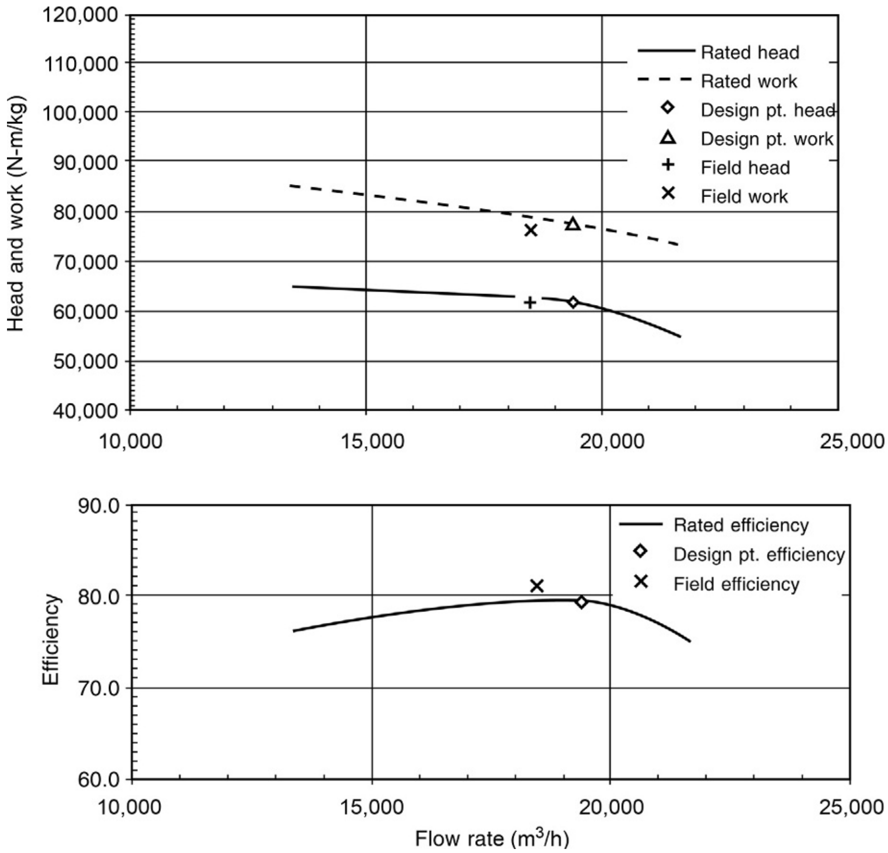


FIG. 8.19 Section 3 of a sidestream refrigeration compressor, Example 8.5. Data have been fan law corrected to design speed conditions.

FIELD DATA ANALYSIS

Trend the performance data (Fig. 7.6) looking for any changes. To confirm accuracy of your data, compare the total compressor power to the driver power and monitor the compressor work input. Work input is usually (but not always) a good indication of the collected data accuracy. Work input values remain constant for varying inefficiencies in the compressor and are generally accurately predicted by the compressor manufacturer. If work input is off design, then there may be instrumentation problems or something affecting the compressor's ability to do work, such as flow swirl.

Plot the manufacturer's data for the compressor head and efficiency versus flow as well as work versus flow. Data plotted on these curves should be fan law corrected to compensate the test data for test speed versus design speed. PTC 10-1997 suggests using dimensionless coefficients (Chapter 2) for best assessing the performance data.

This page intentionally left blank

Chapter 9

Compressor String Analysis

While it is valuable to routinely complete field performance analysis for your compressor, it is equally important to complete an analysis on the driver. The equipment drivers and plant system can dramatically affect the compressor performance and sometimes what on the surface looks like a compressor problem is instead a driver or system issue. Low efficiency or other issues with the driver, condenser, evaporator or catalyst bed issues, or other system problems may be the root cause of what looks like a compressor performance problem; thus, a full string analysis along with a review of any system issues should always be completed if at all possible.

In Chapter 8, the example for the wet-gas compressor showed that the problem was not only the compressor but the steam turbine as well. Both needed attention in order to get production back to normal. Likewise, the air compressor string analysis showed the steam turbine performance was significantly off, thus limiting power output and thus capacity of the compressor string.

In this chapter, other situations are presented, including a gas turbine-driven compressor where the compressor is used to evaluate the gas turbine.

GAS TURBINE DRIVERS

Gas turbines may seem too complicated or overwhelming at first glance, but for regular field monitoring on a relative basis, it is not all that difficult. A regular check on the overall efficiency can tell a lot about the condition of the turbine and help decide if further investigation is necessary. Trending the turbine efficiency over a long period of time can be very helpful in your condition-based monitoring program. All that is needed is the power of the driven equipment and fuel consumption. A little more information can provide a breakdown between the air compressor portion of the gas turbine and the hot-gas expander section to further pinpoint any problems that may be developing over time.

Example 9.1

A gas transmission company has a series of gas turbine-driven compressors to boost gas pressure in a natural gas pipeline system (Fig. 9.1). In order to better schedule maintenance on the equipment, routine efficiency assessments are completed and plotted over time. The gas turbine fuel is taken directly from the natural gas pipeline.

The natural gas centrifugal compressor for this example is made up of two single-stage compressor bodies in series driven by a gas turbine burning pipeline gas similar to the arrangement in Fig. 9.1 (which shows a single compressor). There is no intercooling between the compressors; thus, they are treated as a single multistage compressor.

NATURAL GAS CENTRIFUGAL COMPRESSOR PERFORMANCE

Tables 9.1 and 9.2 note the operating conditions for this compressor along with results. A computer program Gas Flex utilizing BWR equations of state was used to determine the compressor head, efficiency, and power. These values were plotted on the OEM performance curve to show relative performance. As can be seen in Fig. 9.2, the work input is very close to the predicted value; thus, the power for the compressor can be considered to be accurate.

Compressor Performance. https://doi.org/10.1016/B978-0-12-814219-6.00009-X

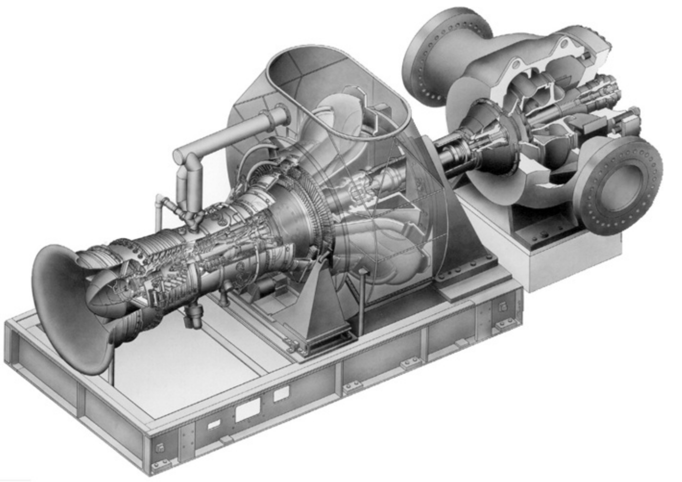


FIG. 9.1 Natural gas compressor driven by a gas turbine. The fuel for the gas turbine is the same as the gas the compressor is moving through the pipeline.

TABLE 9.1 Natural Gas Compressor & Gas Turbine-Driven Operating Data

Ambient		
Atmospheric pressure	psia	14.37
Atmospheric temperature	F	73
Atmospheric humidity	%	79
Compressor		
Inlet pressure	psia	489.11
Inlet temperature	F	64.4
Discharge pressure	psia	1008.3
Discharge temperature	F	185.5
Speed	RPM	4010
Inlet volume flow	cfm	11,134
MW		16.45
Mass flow	lb/min	17,005
Gas power	Horsepower	24,111
Work, adiabatic	ft-lb/lb	46,790
Head, adiabatic	ft-lb/lb	36,497
Efficiency	%	78.0
Gas turbine		
Mechanical losses (Gear + Bearings & Seals)	Horsepower	603
Fuel flow	scfh	190,402
Overall efficiency	%	32.7
Air compressor section		
Inlet pressure	psia	14.1
Discharge pressure	psia	165.3

TABLE 9.1 Natural Gas Compressor & Gas Turbine-Driven Operating Data—cont'd

Discharge temperature	F	689
Air flow rate	scfm	149,667
Head, adiabatic	ft-lb/lb	101,942
Efficiency, adiabatic	%	87.1
Gas power	Horsepower	41,090
Hot-gas expander section		
Inlet temperature	F	2010
Exhaust temperature	F	1044
Flow	scfm	152,840
Power	HP	77,817
Efficiency	%	86.7

TABLE 9.2 Gas Analysis, Mole %

C6+	0.003
Propane	0.134
i-Butane	0.0156
n-Butane	0.1835
i-Pentane	0.00675
n-Pentane	0.005
Nitrogen	1.17
Methane	96.1973
Carbon dioxide	0.75
Ethane	1.7
MW	16.45
Heating value	1011.2
Compressibility	0.9979

GAS TURBINE OVERALL EFFICIENCY

The gas turbine overall efficiency can be calculated by [36]:

$$\eta = \frac{(BHP)(2544.43)}{Q \times HeatingValue} \tag{9.1}$$

where

η = Overall thermal efficiency of gas turbine
BHP = Brake horsepower of driven equipment (Generator, pump, or compressor power + bearings & seal losses, include bearing and seal losses for gas turbine), horsepower
Q = Fuel flow rate, natural gas, std. cu ft/h
Heating value = Btu/std. cu ft

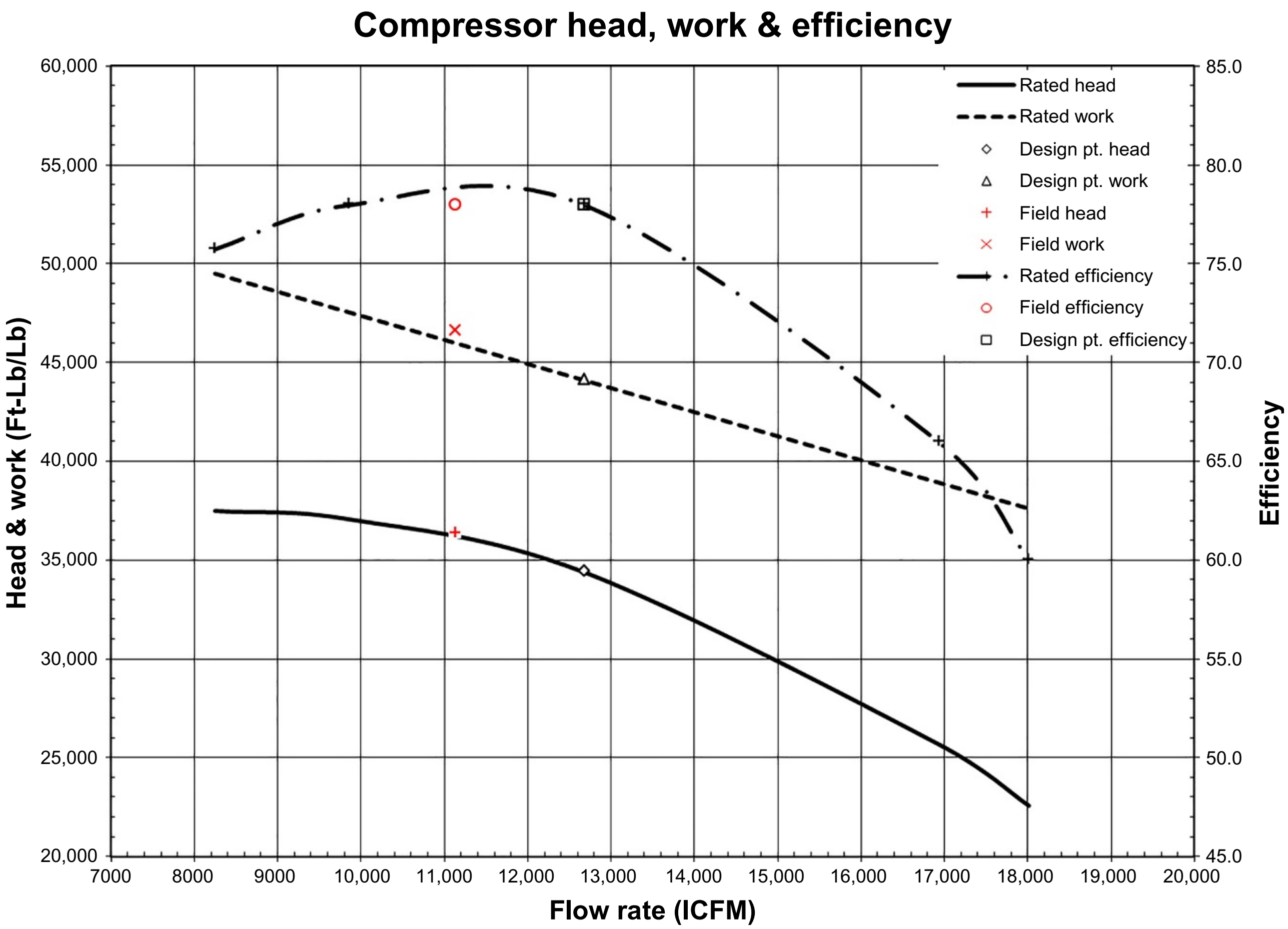


FIG. 9.2 Natural gas compressor performance. Note that the operating data have been corrected for speed using fan laws. Also note that the work input is close to the predicted values; thus, the performance values can be deemed reliable.

Notes—for Pittsburgh Natural Gas [37]:

1. Normal air flow required for combustion of natural gas, 10.62 std. cu ft air/std. cu ft gas.*
2. Net heating value = 1021 Btu/std. cu ft gas
3. 106.2 gross Btu/std. cu ft air
4. Gas volume flow is for gas saturated with water vapor at 60 F and 29.92 in. of mercury absolute.
5. Typical Pittsburgh Natural Gas analysis: CH_4, 83.4%, C_2H_6, 15.8%, N_2, 0.8%—% by volume.
6. Typical exhaust gas from combustion of natural gas: CO, 0.223; CO, 0.228; N_2, 0.535; H_2, 0.014.

*Note: While 10.62 cu ft is the amount of air required for the burning of 1 cu ft of natural gas (primary air), significantly more air is used to keep the engine cool (secondary air). 2, 3, or even 4 times as much air may be used for this purpose.

Overall gas turbine efficiency is calculated using Eq. (9.1):

$$\eta = \frac{(24,111 + 603)(2544.43)}{190,402 \times 1011.2}$$

$$\eta = 32.7\%$$

THERMAL EFFICIENCY OF AIR COMPRESSOR SECTION OF GAS TURBINE

Procedure: Calculate adiabatic head, Eq. (7.5a), Adiabatic Efficiency, Eq. (7.8), and adiabatic power, Eq. (7.13).

$$H_{ad} = RT_1 \left(\frac{k}{k-1} \right) \left[\left(\frac{P_2}{P_1} \right)^{(k-1)/k} - 1 \right] \tag{7.5a}$$

$$\eta_{ad} = \frac{T_1 \left[P_2/P_1 \right)^{(k-1)/k} - 1 \right]}{T_2 - T_1} \tag{7.8}$$

$$\mathrm{GHP}_{ad} = \frac{H_{ad} \times Q_1 \times 144 \times P_1}{\eta_{ad} \times T_1 \times R \times 33000} \tag{7.13}$$

See note #1 to determine air flow. Also, reference example #7.2.

Assuming dry air and using Note #1, the air compressor section results are (see Table 9.1) shown as follows. A computer program Air Flex using the adiabatic equations shown here was used for computing the results.

Flow rate = 149,667 cfm
Head = 101,942 ft-lb/lb
Efficiency = 87.1
Power = 42,090 HP

THERMAL EFFICIENCY OF HOT-GAS EXPANDER SECTION OF GAS TURBINE

The efficiency of the hot-gas expander section can be calculated by[36]:

$$\eta = \frac{(BHP)(2544.43)}{Q \times HeatingValue} \tag{9.2}$$

where

η = Thermal efficiency of hot-gas expander section of gas turbine
BHP = Brake horsepower of driven equipment (Generator, pump, or compressor power + bearings & seal losses, include bearing and seal losses for gas turbine) + the power of the air compressor section of the gas turbine, horsepower
Q = Fuel flow rate, natural gas, std cu ft/h
Heating Value = Btu/std. cu ft

If upstream and exhaust pressures and temperatures are available, use a hot-gas expander calculation. For hot-gas flow rate, consider note #1 above for estimating air flow. Add air flow to fuel flow to determine total flow through the hot-gas expander.

Note the typical gas analysis of a hot-gas expander is: O_2, 1.0%, N_2, 71.0%, H_2O, 11.0%, CO_2, 17.0% [37]

Using Eq. (9.2), the efficiency for this expander is:

$$\eta = \frac{(24,111 + 603 + 41,090)(2,544.43)}{190,402 \times 1,011.2}$$

$$\eta = 86.96\%$$

Using the BWR gas properties program Gas Flex, an efficiency of 86.96% was calculated for the expander section of the gas turbine.

MOTOR DRIVER

On the surface, a string analysis involving a gas compressor (Fig. 9.3) and motor driver is relatively simple; however as shown in this example, it is important to always have a means of verifying results. Question everything including things you normally assume to be valid, accurate values.

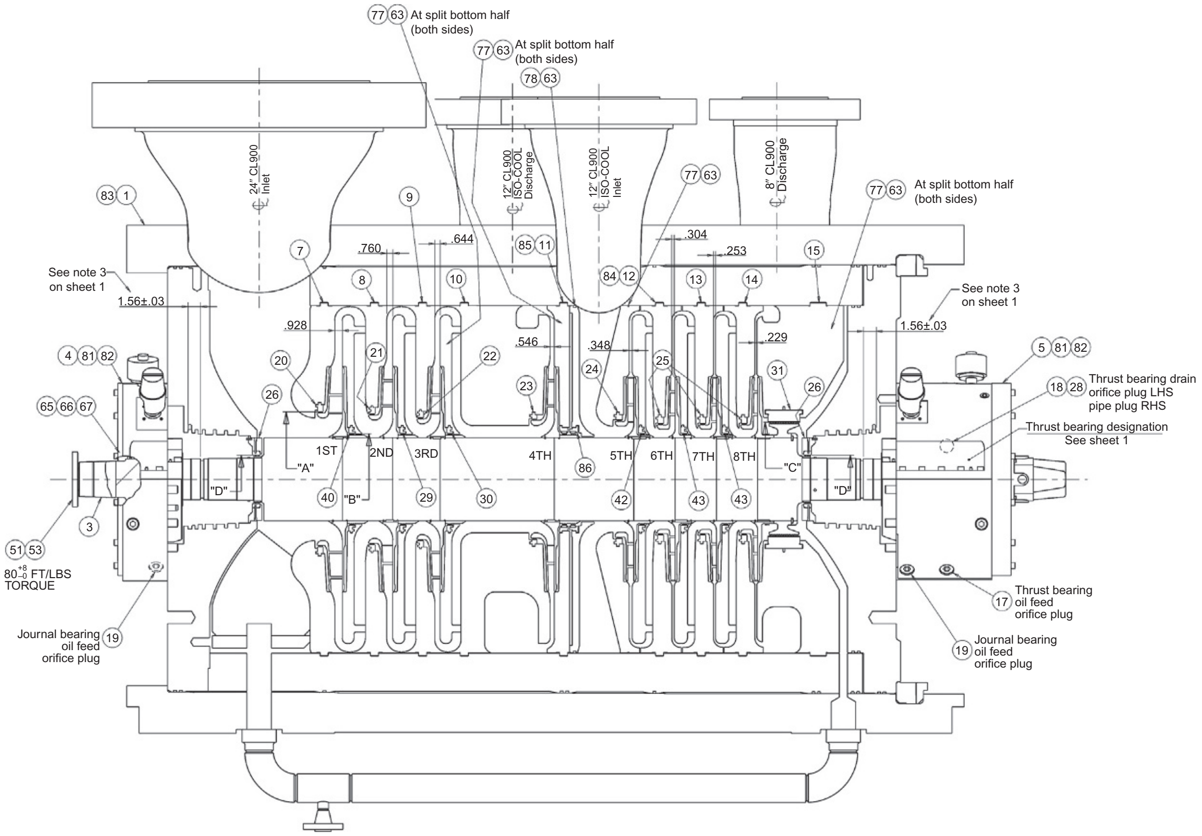


FIG. 9.3 Isocooled compressor. The client reported the compressor power to be high and the volume flow low. *(Compressor cross section courtesy Elliott Co., Jeannette, PA, USA.)*

Example 9.2

This example involves a situation where the compressor power was much higher than design values based on motor power calculations. The compressor volume flow rate was reported to be very low for the amperage draw. An analysis of the compressor performance was made, but all indications were that the compressor power was in spec (Figs. 9.4 and 9.5).

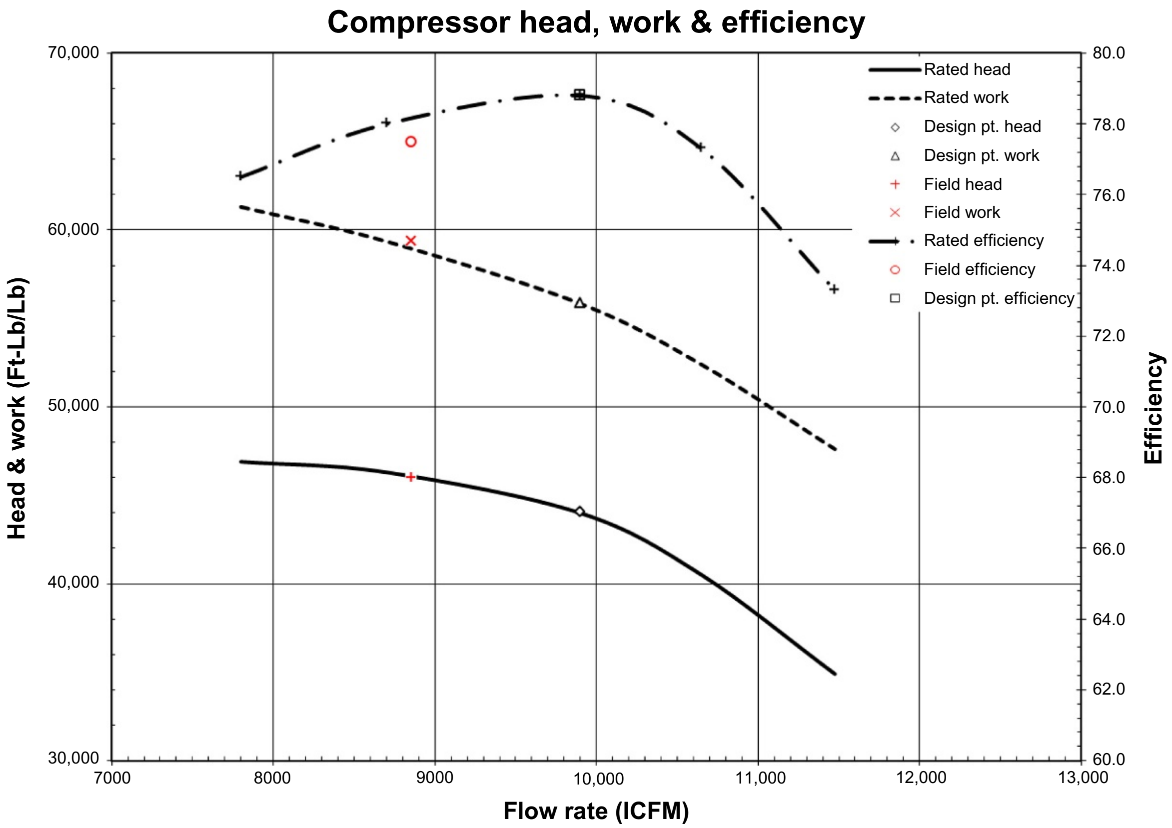


FIG. 9.4 Section 1 performance. Note that while the efficiency is slightly low, the head and work input are right on. Since the work input is per predicted values, the calculated gas power can be assumed to be accurate.

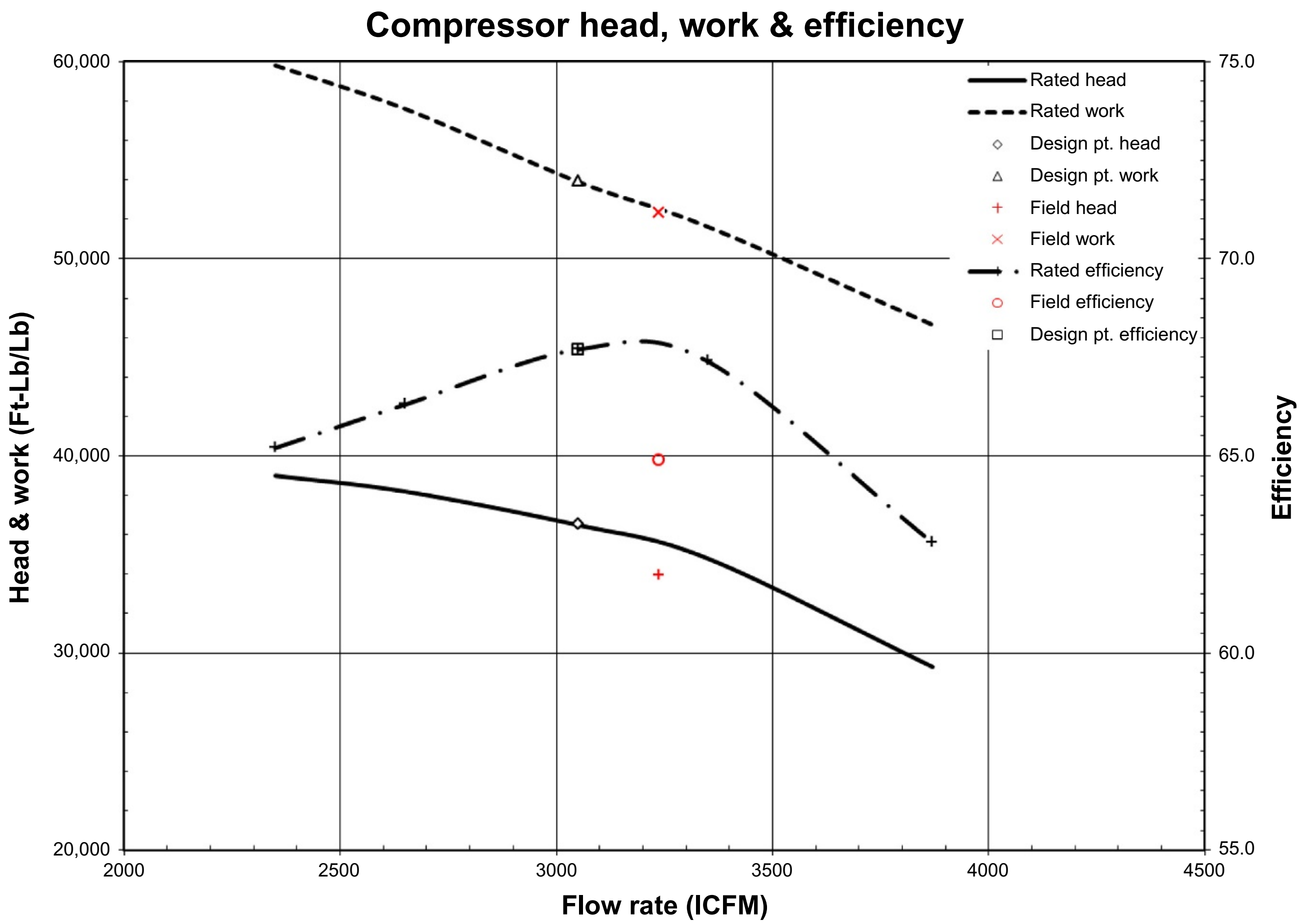


FIG. 9.5 Section 2 performance. Note that while the efficiency and head are low, the work input is right on. Since the work input is per predicted values, the calculated gas power can be assumed to be accurate.

Tables 9.3 and 9.4 summarizes the operating data and results from Gas Flex, a compressor program that uses modified BWR equations of state. A power balance was made to confirm the power calculated by the compressor performance calculations agree with the motor amperage calculations.

$$\begin{aligned}\text{Total compressor gas power} &= 7690 + 6788\,\text{HP} \\ &= 14{,}478\,\text{HP}\end{aligned}$$

Motor power is calculated by:

$$\begin{aligned}\text{BHP} &= E \times I \times \eta \times P.F. \times \sqrt{3}/746 \\ &= 6366 \times 1331 \times 0.96 \times 0.92 \times 1.732/746 \\ &= 17{,}375\,\text{HP}\end{aligned}$$

Power Balance:

Mechanical losses were estimated to be 498 HP (1% for bearings and seals and 2% for the gear)

$$\begin{aligned}\text{Test Error} &= \left[\left(\frac{\text{Driver Power} - \text{Losses}}{\text{GHP}}\right) - 1\right] \times 100\% \\ &= \left[\left(\frac{17{,}375 - 498}{14{,}478}\right) - 1\right] \times 100\% \\ &= 16.6\%\end{aligned}$$

Since the power balance was so far off, head and efficiency were also calculated with another program using another software package (RKMOD—Redlich-Kwong Modified) to assure accuracy of the Gas Flex calculations. Results were

TABLE 9.3 Gas Analysis

Gas Name	Mole Fraction
1 Butyne	0.00103697
Hydrogen	0.2059345
Carbon monoxide	0.00075206
Methane	0.40608006
Nitrogen	0.00411768
Oxygen	0.00101684
Acetylene	0.1713822
Ethylene	0.01748757
n Pentane	0.00510432
cis 2 butene	0.00101684
trans 2 butene	0.00204374
n Butane	0.02353825
iso Butane	0.01312826
Ethylene oxide	0.03979756
Propane	0.10143193
Ethane	0.00204374
iso Butene	0.00408748

TABLE 9.4 Compressor-Operating Data

Item	Units	Section 1	Section 2
Inlet pressure	psia	133.7	352.2
Inlet temperature	F	107.4	110.9
Discharge pressure	psia	369.5	769.0
Discharge temperature	F	256.0	245.0
Speed	RPM	6780	6780
Inlet volume flow	cfm	8915	3269
MW		21.293	21.293
Mass flow	lb/min	4278.2	4278.2
Gas power	Horsepower	7690	6788
Work, polytropic	ft-lb/lb	59,316	52,359
Head, polytropic	ft-lb/lb	46,014	33,967
Efficiency	%	77.57	64.9

within 1%. Considering the calculated work input values in Figs. 9.4 and 9.5 were close to the predicted values, the conclusion was that the compressor gas power was most likely accurate.

This turned attention to the motor power and what might be wrong. After considerable efforts, the ammeter was found to be in error. Once the ammeter was replaced, the power balance was found to be within 2%.

CONDENSING TURBINE

Example 9.3

Hydrogen Recycle Compressor Field Performance Analysis

As part of a debottlenecking procedure, a refinery in Oklahoma was interested in analyzing its hydrogen recycle compressor performance thinking it may need to be rerated in order to improve plant capacity (Fig. 9.6).

FIG. 9.6 Oklahoma Refinery hydrogen recycle compressor.

The compressor string consisted of a small barrel compressor and a condensing steam turbine driver. Of primary concern for data accuracy were the flow meters and obtaining an accurate gas analysis. Accurate data were a special concern since there is no confirmation of calculation results by completing a power balance. Since the steam turbine is a condensing unit, the turbine power cannot be assessed directly and the turbine power is assumed to be equal to the compressor power it is driving.

Special care was taken to assure accurate data. Pressure differential data from the compressor gas flow meter was read directly and was used to calculate the flow rate to the compressor. The same was done for the turbine flow meter. Multiple gas samples were taken to assure redundancy.

This test had to be repeated several times before satisfactory results were obtained. After completing the first test, the plotted data showed a significant deviation between the compressor work input and the actual calculated value. Accordingly, the results were thrown out and the test repeated until the work input was within reasonable proximity to the predicted work input.

The problem was in the gas-sampling technique. While the bulk of the gas is hydrogen, there are small quantities of heavies in the gas like hexane, propane, and butane. Just a small fractional error in the measurement of these heavies can upset the final mixture and have significant effects on efficiency and power values. It is necessary to completely purge the testing container, but also to heat and insulate it. Then, before running the gas through the gas chromatograph, the gas must be heated again to assure any condensation has been vaporized. Otherwise, a small fraction of the heavies can condense on the container walls and the sample is in error.

The compressor was determined to be operating at about 71% efficiency, about 5 percentage points below predicted values (Fig. 9.7 and Table 9.5).

The turbine was operating at about 44% efficiency, about 10 percentage points below its predicted value.

While the data showed that the equipment needed maintenance to get it back to design-operating condition, the data also revealed that the compressor was operating at midrange and there was no need for a rerate saving the refinery a lot of money, time, and effort.

A lot can be learned from a field performance test. Knowing where the compressor is operating on the curve is valuable information. In this case, it was found that the compressor was not the bottleneck and a rerate was not necessary as originally expected (Table 9.5).

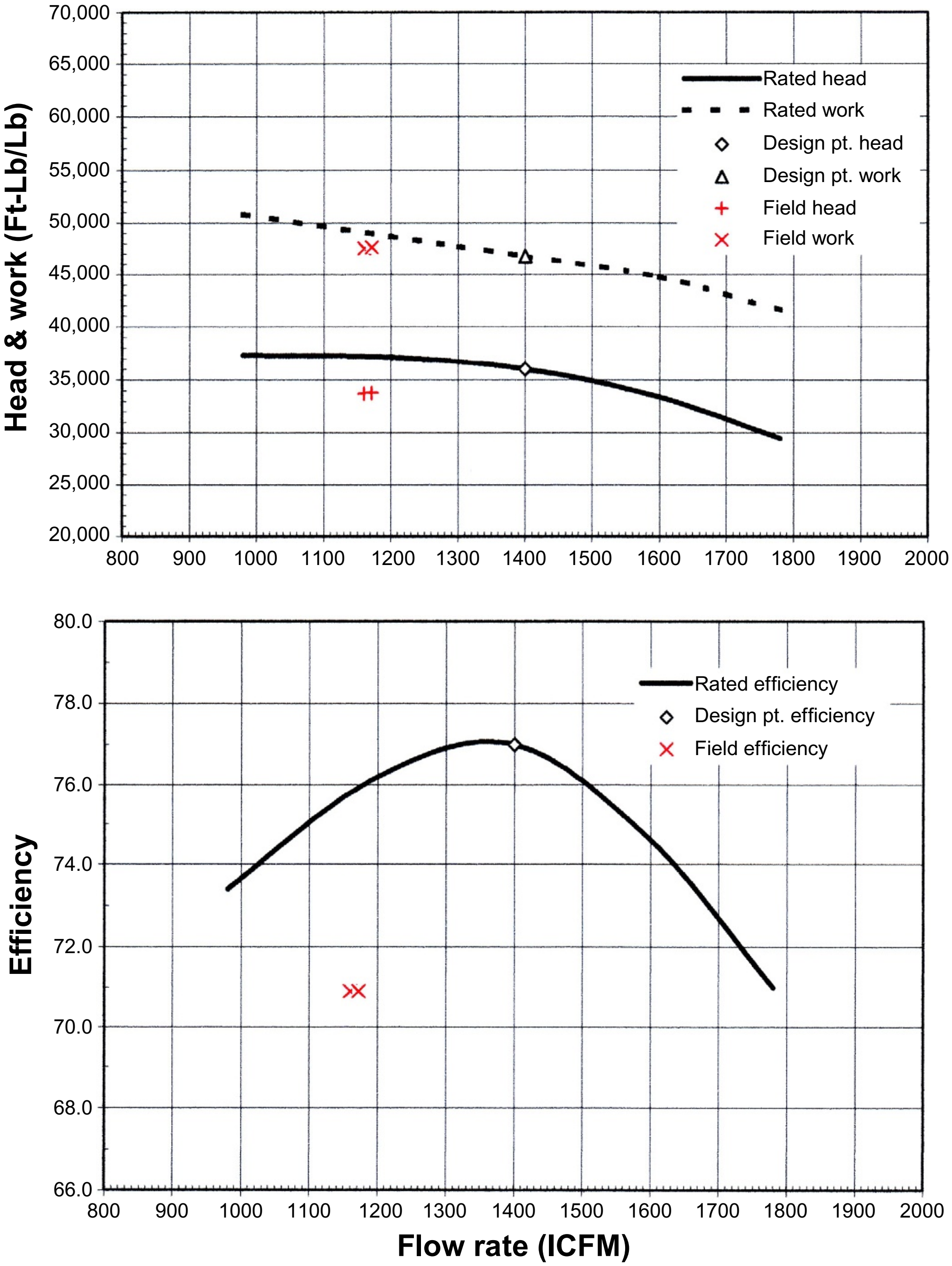


FIG. 9.7 Hydrogen recycle compressor-operating data for August 6, 2004. Data have been fan law corrected to design speed, 11,070 RPM. Accuracy of test was accomplished by matching work input with OEM performance curve.

LIQUID INGESTION

Liquid ingestion can cause several problems with a compressor. Erosion and corrosion can occur and even failure of the internal parts, in particular the impellers (Fig. 4.11). Also, regarding performance analysis, the operating data cannot be used directly. Consideration for the increased density and evaporation effects must be made to conduct an accurate analysis. Liquid entrained in the gas will make the gas-liquid mixture much more dense than the dry gas itself; thus, more compression will occur than for a dry gas. Note also that the calculation assumes a dry gas and the heat of compression will cause some of the liquid to evaporate, thus affecting the measured discharge temperature. If the amount of the liquid is known, then accommodations in the calculation procedure can be made for this; however, if the amount is unknown, it is not possible to accurately calculate the head and efficiency.

TABLE 9.5 Summary of Results Hydrogen Recycle-Operating Data for August 6, 2004

Compressor Data			Gas	Mole Fraction	Formula
Time	7:30 AM	10:30 AM	Hexane	0.0002	C_6H_{14}
Flow, MMSCFD	134	132	Hydrogen	0.92242	H_2
Orifice DP, "H_2O"	19.65	18.9	Propane	0.00346	C_3H_8
Inlet *T*, F	112.5	114	*i* Butane	0.0003	C_4H_{10}
Inlet *P*, psia	1721	1724	*n* Butane	0.00051	C_4H_{10}
Disch *T*, F	143	144	Ethane	0.01788	C_2H_6
Disch *P*, psia	1962	1961	Nitrogen	0.0064	N_2
Speed, RPM	11,289	11,289	Methane	0.04883	CH_4
			MW = 3.5766		
Compressor results					
Flow, #/min	1139	1112			
Flow ICFM	1195	1183			
Head, ft-lb/lb	35,118	35040			
Efficiency	70.9	70.9			
Power, HP	1710	1665			
Turbine data					
Inlet *P*, psig	588	589			
Inlet *T*, F	625	625			
Exh *P*, "hg Vac"	28.5	28.5			
Exh *T*, F	75	75			
Flow, k#/h	19.7	19.55			
Orifice DP, "H_2O"	10.9	10.55			
Turbine results					
Flow #/h	21,568	21,240			
Efficiency	44.3	43.8			
Power, HP	1710	1665			

Note that the compressor power is identical to the steam turbine power. The compressor power was used as input data for the steam turbine calculation to determine the exhaust conditions of the condensing steam turbine. Accuracy of test was accomplished by matching work input with OEM performance curve.

A recycle compressor at an ethylene oxide plant was experiencing some off performance. Continuous remote performance monitoring of the compressor using Gas Flex to establish a long-term trend was initiated to help assess the situation.

The curve in Fig. 9.8 was taken from the OEM curves provided with the API data sheets. The design point from the API data sheet was then run through Gas Flex® and plotted on the performance curve to confirm accuracy. Data for several points in time were plotted on the curve. Motor power was not in agreement with the compressor gas power.

Some points to note:

- The flow rate has been decreasing over the remote-monitoring time period.
- The work input is high for all points and suggests errors in the gas analysis or operating data or possibly liquid ingestion.
- Power balance between the motor and compressor is off.
- The deviation of calculated head versus predicted is high and this deviation appears to increase with increased flow rate.

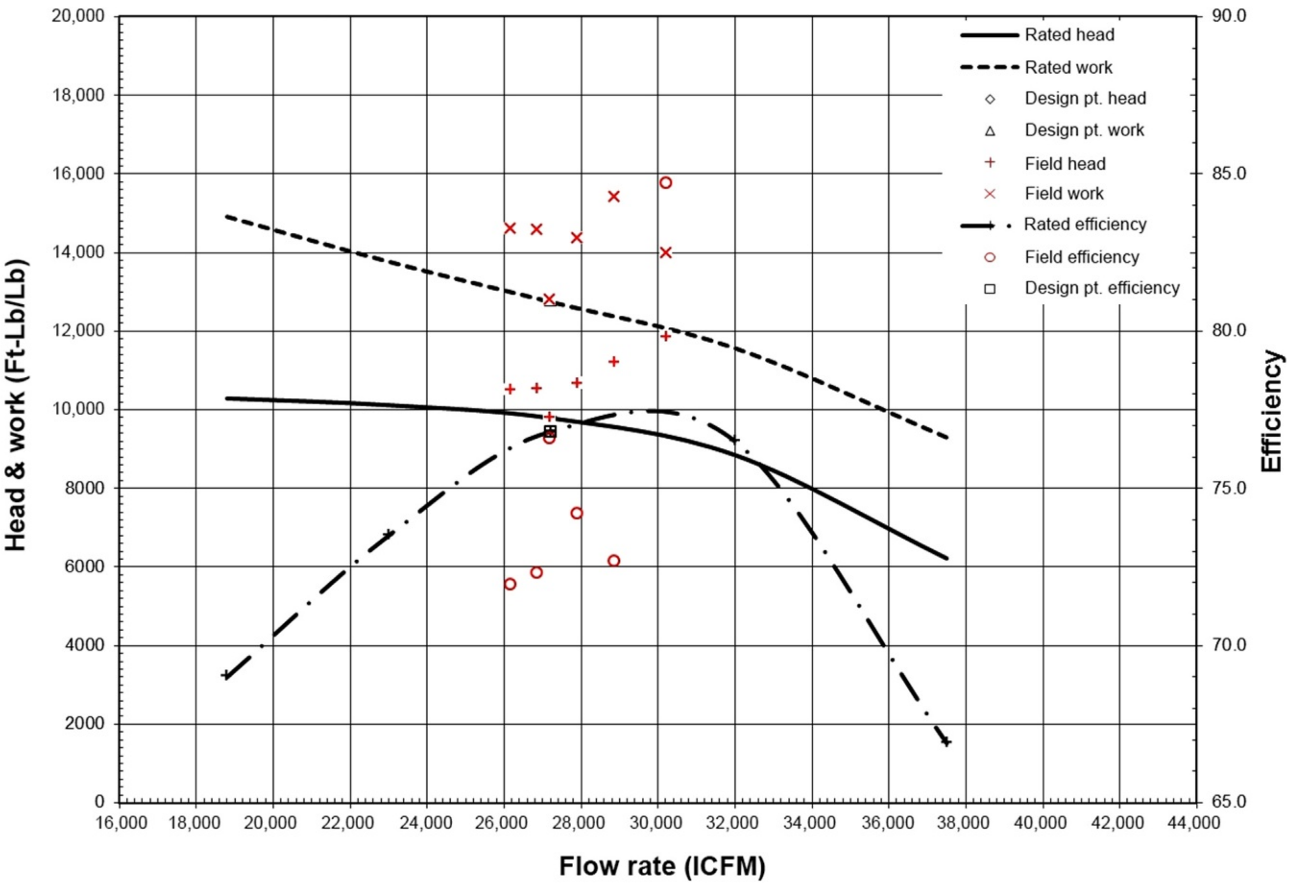


FIG. 9.8 Plot of head, efficiency, and work versus flow. Data corrected by fan laws to the design speed of 8204 RPM.

The original conclusion assumed primarily instrumentation or gas analysis issues as the power balance was off. Discarding the high flow data point (31,000 cfm) due to the unrealistic high efficiency was considered. However, after further analysis, it was confirmed that water ingestion explained the performance deviation and the high efficiency at the 31,000 cfm point.

The compressor was modeled aerodynamically using the Flexware® aerodynamic design software system CompAero. With design point values, the resulting performance closely matches the OEM predicted curve.

To simulate the expected water ingestion, the mole weight of the gas was increased from 24 to 26 MW, and the predicted performance curve was recreated using the denser gas and the actual operating inlet conditions (vs. design point inlet conditions). The new curve (Fig. 9.9) passes through one of the test points and indicates that an 8.3% higher MW gas increases the head (and pressure ratio) by about 6%. This explains the high head. As the flow increases, more water is swept up with the gas, thus explaining the higher head as flow increases.

The highest flow point in Fig. 9.8 (31,000 cfm) was originally considered just bad data; however, this point is most likely affected by the evaporative effect of reducing the measured discharge temperature, thus showing an abnormally high efficiency.

The high head shown in Figs. 9.8 and 9.9 is due to the liquid entrainment and higher than spec density of the gas-liquid mixture.

Liquid carryover of up to 9% by weight (287 gpm) was confirmed to explain the deviation in head by modeling the compressor and producing new prediction curves with entrained water (Fig. 9.9). The continuous remote monitoring is shown in Fig. 9.10.

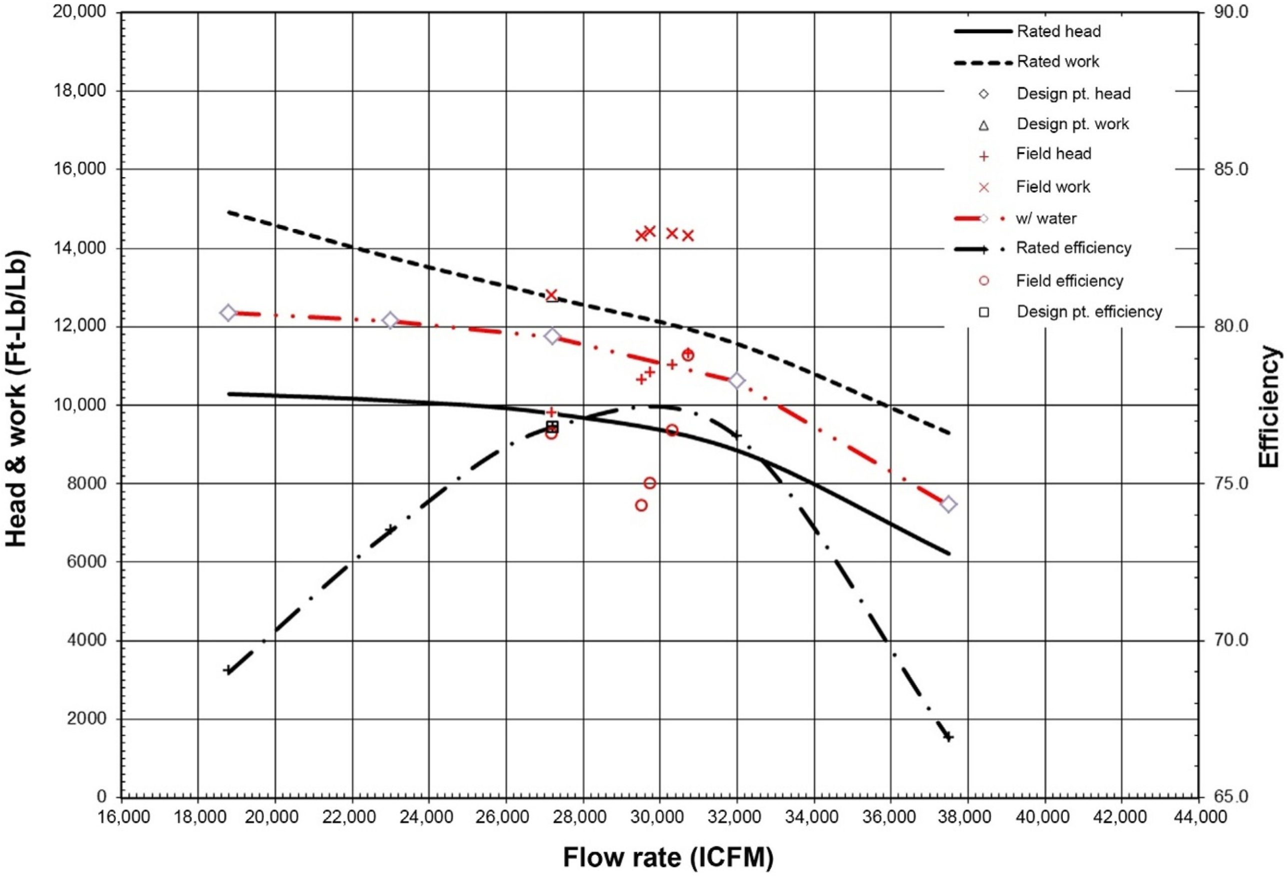


FIG. 9.9 This chart shows the effect of 287 gpm of liquid water ingestion (8.7% by weight) and closely matches higher flow data (head curve ⋄ @ 30100 cfm). Varying the amount of liquid will raise or lower the head curve accordingly. Note that the design points are shown as proof and the model is correct. Other points are actual operating data.

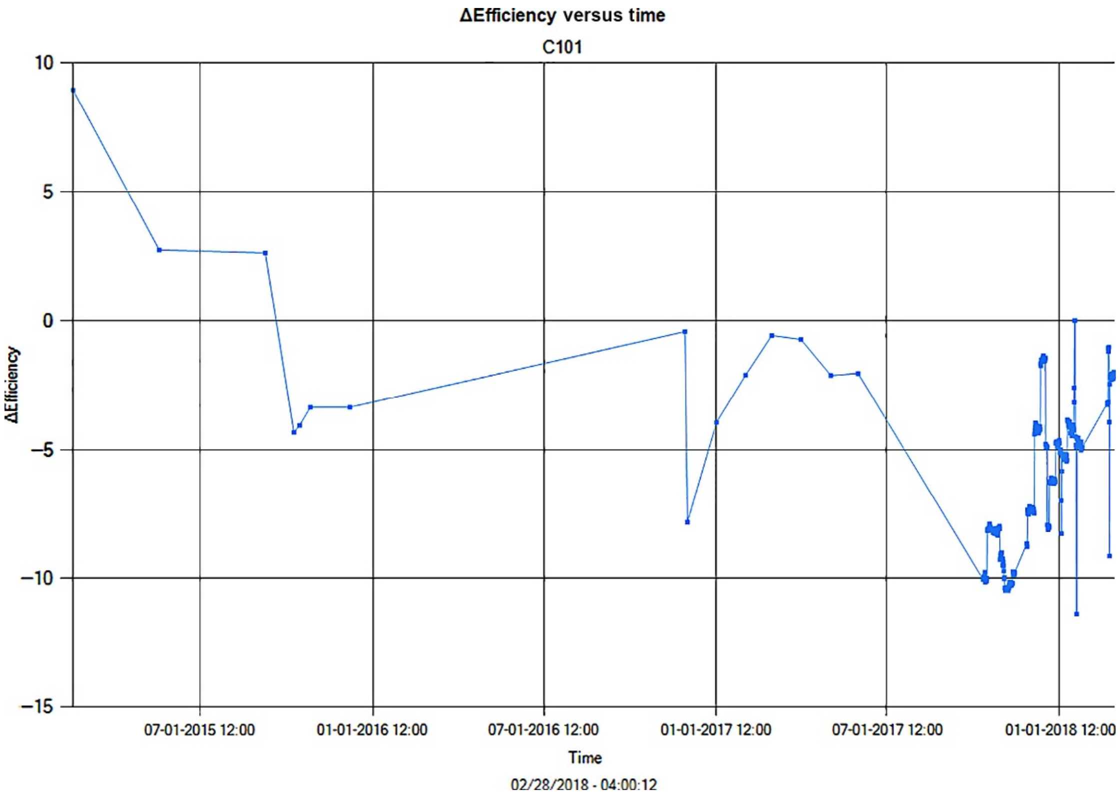


FIG. 9.10 Trend plot of delta efficiency (actual operating efficiency vs. OEM predicted curves). Note that flow had been dropping over time; thus, liquid entrainment also is reduced over time. High flow at initiation of time series chart shows a high efficiency due to liquid evaporation. The most recent data (right of chart) show the more accurate relative efficiency (about 10 points below expected) as flow is lower and less or possibly no liquid is being drawn into the compressor.

This page intentionally left blank

Chapter 10

Flow Meters

Orifice and nozzle meters utilize the relationship between velocity of a flowing fluid and static pressure to measure flow. This relationship is defined by Bernoulli's equation. Simply put, as flow area is reduced (by adding a flow nozzle or orifice), the velocity must increase to maintain the same mass flow (see Fig. 10.1). Bernoulli's equation says that as the velocity is increased, the static pressure is reduced. From this, the flow equations have been developed (see Chapter 2).

It is most crucial that pressures and temperatures be accurately measured to assure proper flow measurement. One important factor in assuring accurate pressure measurement is that the orifice or nozzle meter be properly oriented in the piping to obtain uniform flow upstream of the orifice. According to ASME "Fluid Meters" [29], the required length of straight run of piping upstream of an orifice is shown in Fig. 10.2 and Table 10.1.

SQUARE-EDGED ORIFICES

$$\dot{M} = (5.983)(K)\left(d^2\right)(\mathrm{Fa})(Y)\sqrt{\frac{h_w}{v_1}} \tag{10.1}$$

$$K = C \times E \tag{10.2}$$

$K =$ flow coefficient. If the orifice has not been individually calibrated, obtain C from Table 10.2 in this section, or from [29]. C is approximately 0.6 for high Reynolds number.
$C =$ discharge coefficient. Obtained from the reference tables or from manufacturer's data.

$$= K/E = K\sqrt{1-\beta^4} \tag{10.3}$$

$E =$ velocity of approach factor, defined as

$$\frac{1}{\sqrt{1-\beta^4}} \tag{10.4}$$

where β is the ratio of throat diameter d to pipe diameter D.

$d =$ diameter of orifice in inches.
$\mathrm{Fa} =$ thermal expansion factor. Obtained from Fig. 10.3 or from [29]. Generally, this number is very close to 1.0 for most compressor suction conditions.
$Y =$ net expansion factor for square-edged orifices. Ratio of flow coefficient for a gas to that for a liquid at the same value of Reynolds number. Obtained from Fig. 10.4 or by equation.

Orifice Meter Expansion Factor

For a square-edged orifice-with upstream static pressure tap, use the following equation, or Fig. 10.4 [29].

$$Y = 1 - \left[\left(0.41 + 0.35\beta^4\right)\left(\frac{P_2 - P_1}{P_1}\right) \div k\right] \tag{10.5}$$

Compressor Performance. https://doi.org/10.1016/B978-0-12-814219-6.00010-6

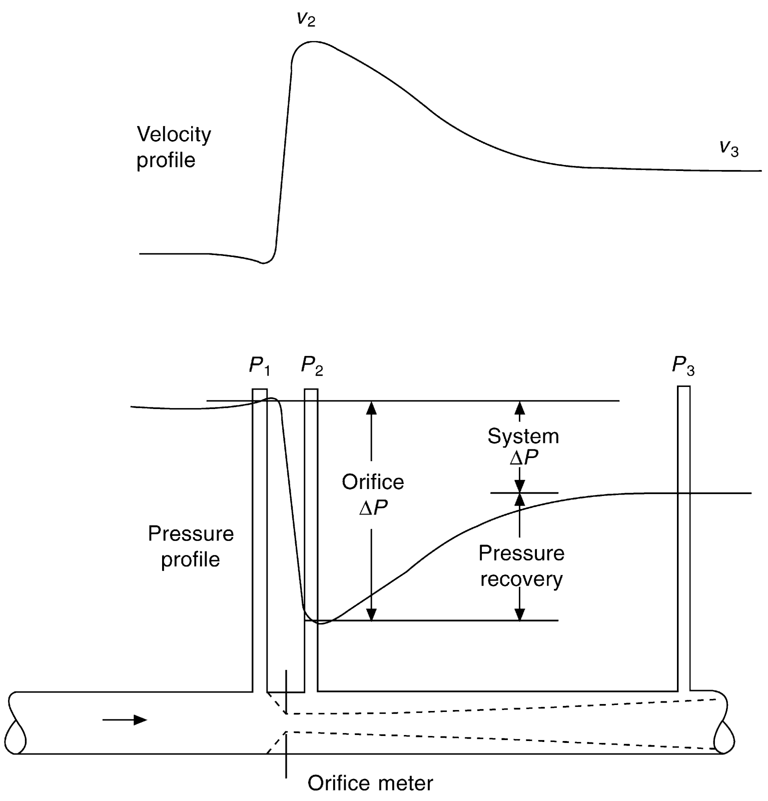


FIG. 10.1 Orifice meter. *(Data from The orifice meter. Pittsburgh, PA; Rockwell Manufacturing Co.; 1938.)*

Orifice Discharge Coefficient

For best results, the orifice meter should be calibrated to determine the discharge coefficient. If this is not possible, Table 10.2 may be used.

FLOW NOZZLES AND VENTURI TUBES[1]

$$\dot{M} = (5.983)(C)(E)\left(d^2\right)(\text{Fa})(\text{Ya})\sqrt{\frac{h_w}{v_1}} \tag{10.6}$$

C = discharge coefficient. Approximate value for flow nozzles is 0.98 for high Reynolds number (see Table 10.3).
E = velocity of approach factor
$= \frac{1}{\sqrt{1-\beta^4}}$
d = diameter of venturi or nozzle throat in inches
Fa = thermal expansion factor (see Fig. 10.3).
Ya = expansion factor for flow nozzles or venturi tubes (see Fig. 10.5).

1. Adapted from "Fluid Meters," ASME PTC 19.5, 1971 [29].

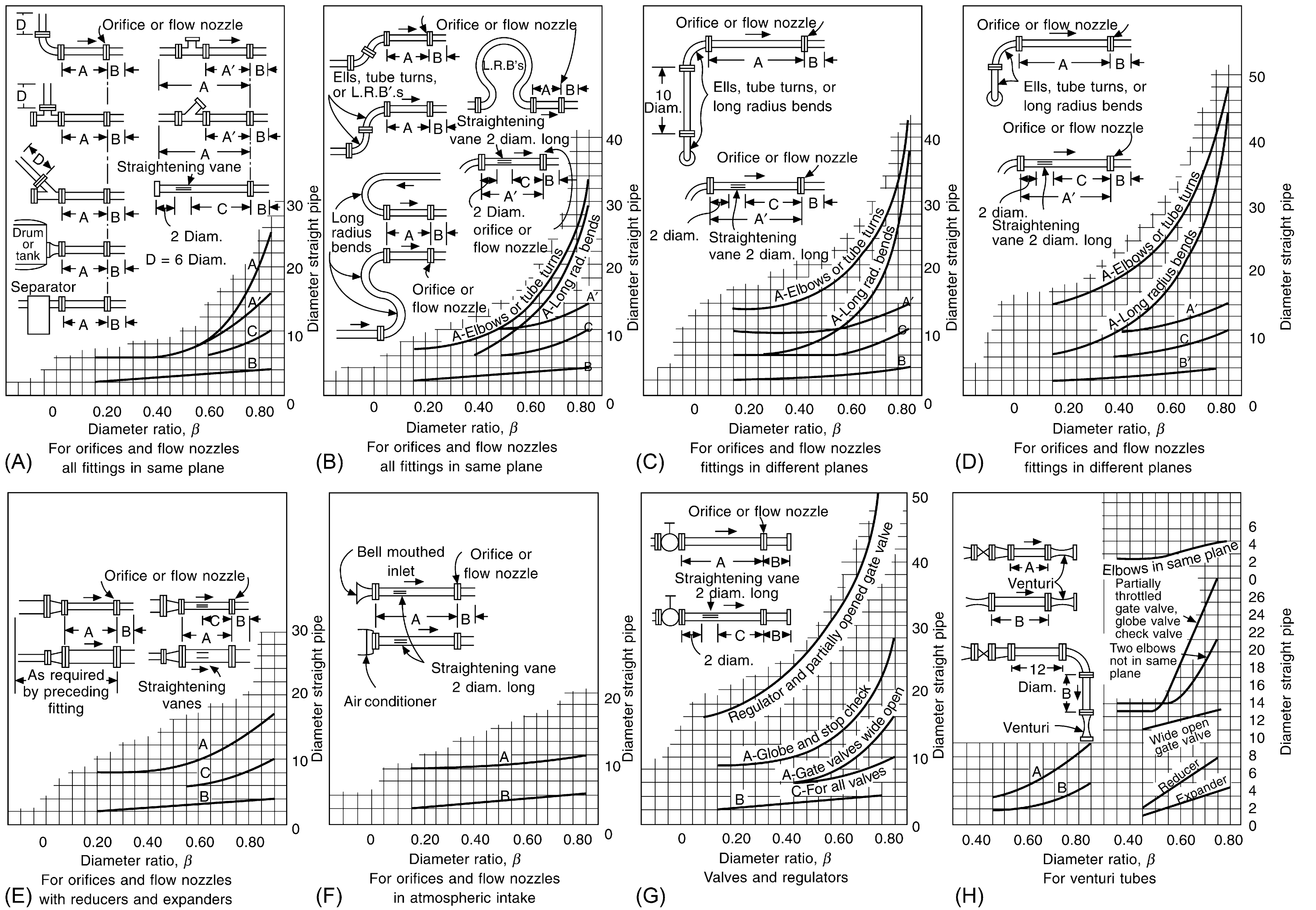


FIG. 10.2 Recommended minimum lengths of pipe preceding and following orifices, flow nozzles, and venturi tubes (all control valves, including regulators, should be located on outlet side of primary element). *(Data from* Fluid meters. *ASME PTC 19.5. New York: American Society of Mechanical Engineers; 1971.)*

TABLE 10.1 Required Straight Run of Piping for Orifice

Upstream Configuration	Diameters of Straight Run Required	
	$B = 0.5$	$B = 0.8$
Long-radius elbow	7	20
Two elbows—same plane	8	23
Two elbows—different plane	12	32
Partially closed valve	25	50
Full-open gate valve	6.5	13.5
Check valve	11	21

TABLE 10.2 *C*, **Coefficient of Discharge for Square-edged Orifices with Flange Taps**[a]

Reynolds Number	50,000	100,000	500,000	1,000,000
β	**16″ Pipe**			
0.5	0.6197	0.6110	0.6039	0.6031
0.6	0.6362	0.6197	0.6065	0.6049
0.7	0.6548	0.6270	0.6047	0.6020
0.75	0.6619	0.6276	0.6001	0.5966
	2″ Pipe			
0.5	0.6102	0.6076	0.6055	0.6052
0.6	0.6158	0.6118	0.6086	0.6082
0.7	0.6183	0.6124	0.6077	0.6071
0.75	0.6230	0.6160	0.6104	0.6097

[a]For high Reynolds numbers, a factor of 0.6 can be assumed and will be within ±2%. This is true for *D* and $\frac{1}{2}$ *D taps, and vena contracta taps.*
From *Fluid meters*. ASME PTC 19.5. New York: American Society of Mechanical Engineers; 1971.

Flow Nozzle Expansion Factor

For flow nozzles or venturi tubes, use the following equation or Fig. 10.5 [29].

$$\mathrm{Ya} = \left[\left(r_p\right)^{2/k} \times \left(\frac{k}{k-1}\right) \times \left(\frac{1-\left(r_p\right)^{(k-1)/k}}{1-r_p}\right) \times \left(\frac{1-\beta^4}{1-\beta^4\left(r_p\right)^{2/k}}\right)\right]^{1/2} \tag{10.7}$$

Flow Nozzle Discharge Coefficient

For best results, the flow meter should be calibrated to determine the discharge coefficient. If not, however, the following may be used.

For long-radius flow nozzles with pipe taps at 1 diameter and $\frac{1}{2}$ diameter, use the following equation or Table 10.3 [29].

$$C = 0.99622 + 0.00059D - \left(6.36 + 0.13D - 0.24\beta^2\right)\frac{1}{\sqrt{\mathrm{Re}}} \tag{10.8}$$

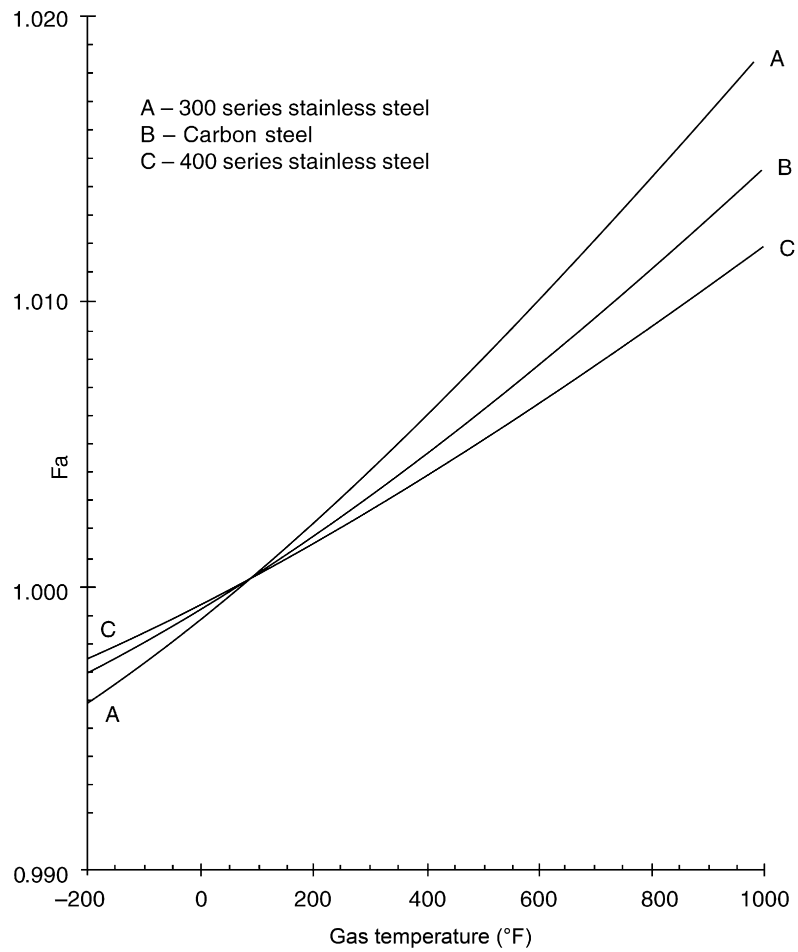


FIG. 10.3 Thermal expansion factor, Fa. *(Adapted from* Fluid meters. *ASME PTC 19.5. New York: American Society of Mechanical Engineers; 1971.)*

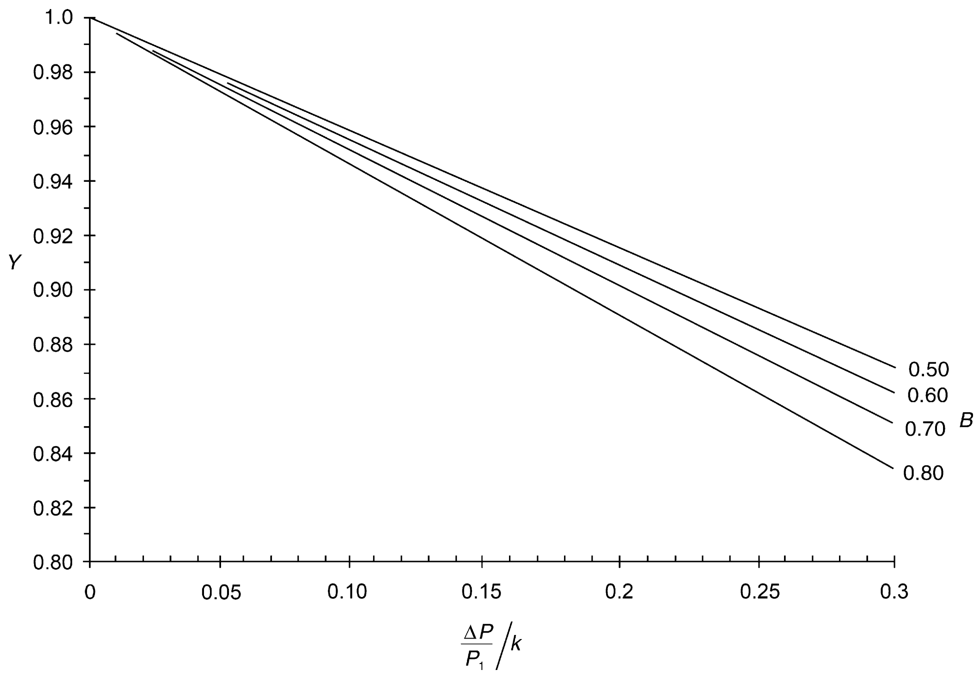


FIG. 10.4 Orifice expansion factor, Y. Expansion factor for thin-plate square-edged orifice with flange taps, D and $\frac{1}{2}D$ taps, or vena contracta taps. Static pressure measured upstream of orifice. *(Data from* Fluid meters. *ASME PTC 19.5. New York: American Society of Mechanical Engineers; 1971.)*

TABLE 10.3 Discharge Coefficient for Long-Radius Flow Nozzles, C[a]

Reynolds Number	50,000	100,000 500,000	1,000,000
β	16″ Pipe		
0.5	0.9681	0.9790 0.9935	0.9969
0.6	0.9683	0.9791 0.9935	0.9969
0.7	0.9684	0.9792 0.9936	0.9970
0.75	0.9685	0.9792 0.9936	0.9970
	2″ Pipe		
0.5	0.9681	0.9767 0.9881	0.9909
0.6	0.9682	0.9767 0.9882	0.9909
0.7	0.9683	0.9768 0.9882	0.9909
0.75	0.9684	0.9769 0.9883	0.9909

[a]Note that a discharge coefficient of 0.98 can be assumed for high Reynolds number and accuracy will be within 1%. For venturi nozzles, use 0.984 for Reynolds number of 200,000 or greater.
From *Fluid meters*. ASME PTC 19.5. New York: American Society of Mechanical Engineers; 1971.

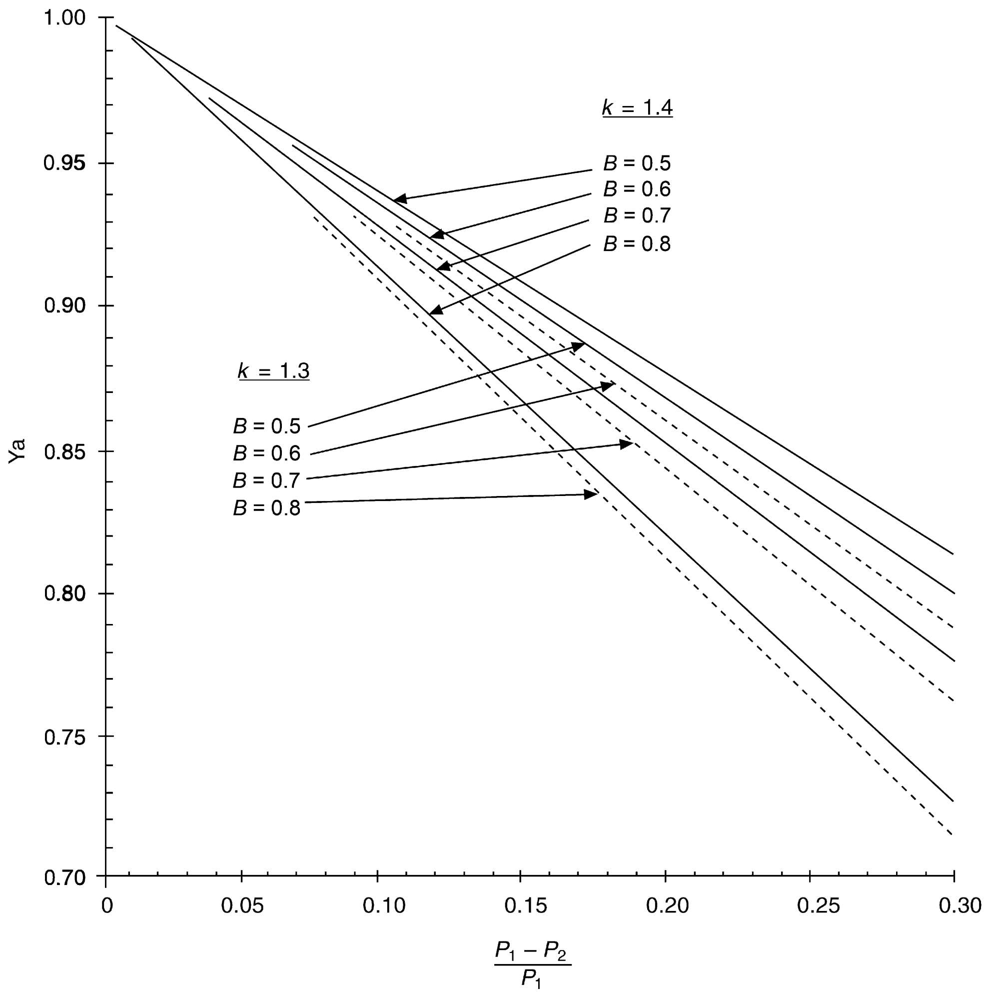


FIG. 10.5 Expansion factor for flow nozzles and venturi tubes, Ya. *(Data from* Fluid meters. *ASME PTC 19.5. New York: American Society of Mechanical Engineers; 1971.)*

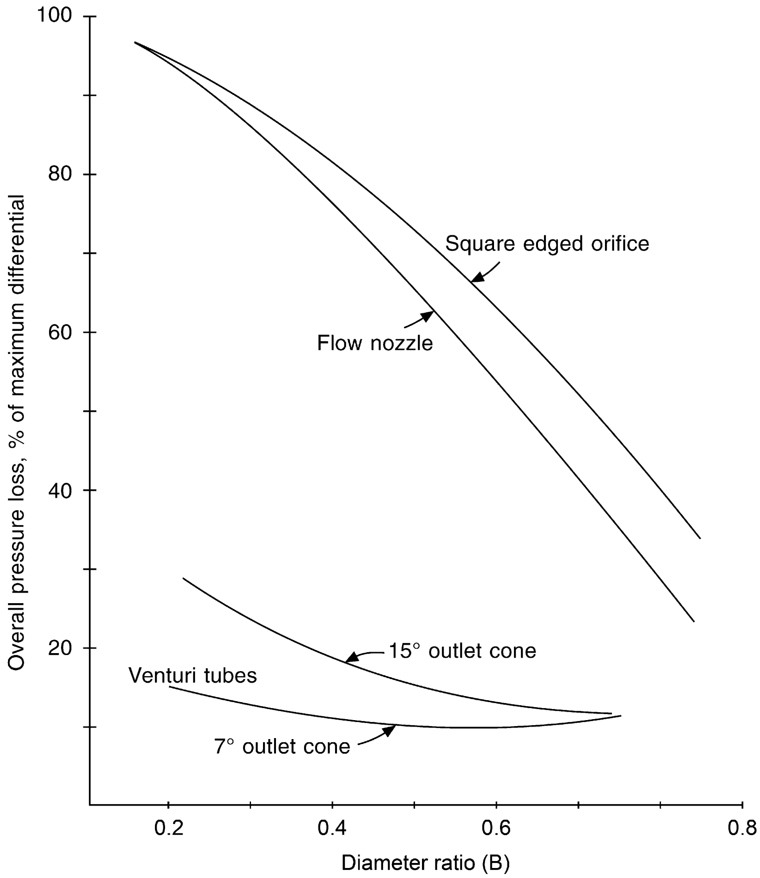


FIG. 10.6 Recovery factor. *(Data from* Fluid meters. *ASME PTC 19.5. New York: American Society of Mechanical Engineers; 1971.)*

RECOVERY FACTOR

If the pressure were measured in several pipe diameters upstream and downstream of the orifice meter or flow nozzle, the pressure difference would be found to be much less than the pressure drop measured for flow measurement. This reduced pressure drop is due to the recovery factor and is the pressure drop to be used when determining the resistance of the flow meter (Fig. 10.6; see also Fig. 10.1).

PITOT TUBE

The pitot tube can be a very accurate means of flow measurement if properly done. The process pipe cross-section should be divided up into several equal area sections, and a midpoint total pressure reading should be taken for each area. Preferably this is done in each quadrant to compensate for nonuniform flow (see Fig. 10.7). Static pressure is measured at a normal pressure tap at the pipe wall.

These pressure measurements are then converted to velocity data.

$$P_{\text{total}} = P_{\text{static}} + P_{\text{velocity}} \tag{10.9}$$

The pitot tube measures stagnation or total pressure.

$$P_{\text{vel}} = P_{\text{total}} - P_{\text{static}}$$

$$V = \sqrt{2g_c(144)vP_{\text{vel}}} \tag{10.10}$$

After determining the fluid velocity at each equal area, the average velocity is then used to determine the flow rate.

$$Q = 60VA \tag{10.11}$$

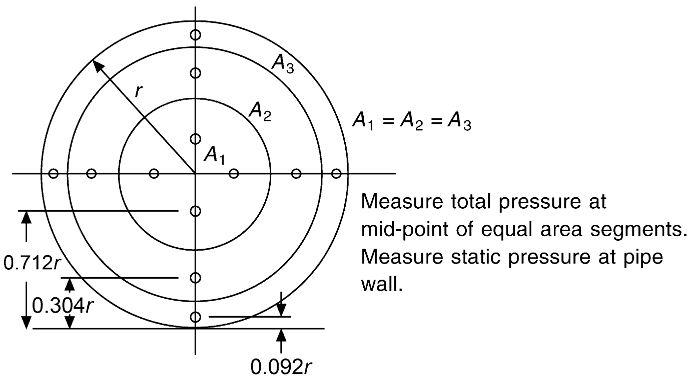


FIG. 10.7 Pitot tube traverse locations.

where

Q = flow rate, CFM
A = pipe area, sq ft
$g_c = 32.2$ ft/s^2
v = specific volume
V = velocity, ft/s

Caution: The alignment of the pitot tube with the flow stream is critical. A very small angular misalignment can cause significant error. Special precautions should therefore be taken to assure precision alignment with the flow stream.

ANNUBAR® FLUID FLOW METERS

A very simple device, both to install and to use, is an Annubar® fluid flow meter (Fig. 10.8). Procedures for measuring the flow rate are similar to an orifice plate or venturi meter.

As with any flow meter, the Annubar® fluid flow meter is very sensitive to velocity profile. It is recommended that a very conservative approach be taken when installing the flow meter so there is no effect of upstream flow distortion from elbows, valves, or other devices. Use the same guidelines established for orifice meters and venturi meters (Fig. 10.2). If the straight run upstream of an elbow is questionable, use a vaned elbow (Fig. 6.20) to assure a uniform velocity profile and accurate results. Also be especially cautious of the alignment of the meter to the piping. A very small alignment error can cause a significant flow rate error.

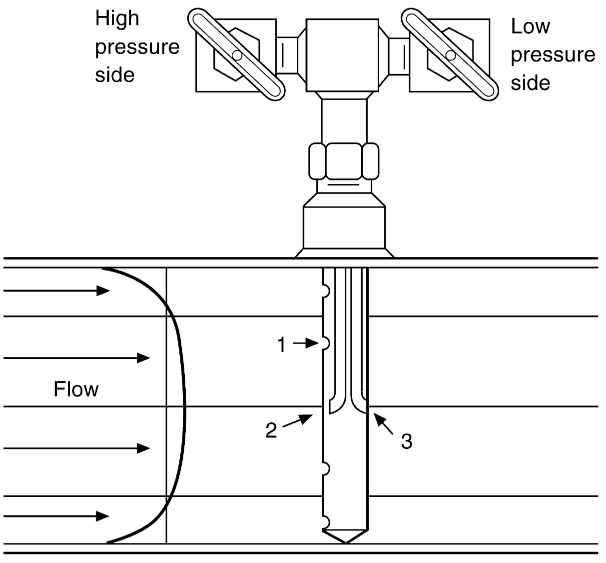


FIG. 10.8 Annubar® flow sensor.

Use the following equation for calculating flow rates or use Flow Flex:

$$\dot{M} = 5.982KD^2 Y_A F_{AA} \sqrt{\frac{h_w}{v}} \tag{10.12}$$

where

K = Annubar® flow coefficient (supplied by manufacturer)
D = pipe diameter, inches
$\dot{M}$ = weight flow, lb/min
Y_A = gas expansion factor
$Y_A = 1 - ((1 - B)^2 \times 0.011332 - 0.00342)\,\frac{h_w}{Pk}$
(Dieterich Standard Diamond Annubar)
$Y_A = 1 - (0.05445 - 0.05703(1 - B)^2)\,\frac{h_w}{Pk}$
(Dieterich Standard Streamlined Annubar)
B = Annubar® blockage = $4d/\pi D$
d = Annubar® diameter, inches
P = pressure of gas at Annubar®, psia
k = ratio of specific heats
F_{AA} = thermal expansion factor. See Appendix E.
h_w = differential pressure, inches water
v = specific volume, ft³/lb

Example 10.1

Steam flowing at 500 psia and 620°F in a 24-in. diameter pipe has a 15-in. water column differential pressure across the Diamond Annubar® fluid flow meter. Calculate the steam flow rate in lb/min as shown below or use Flow Flex for quicker results.

$$\dot{M} = 5.982KD^2 Y_A F_{AA} \sqrt{\frac{h_w}{v_1}}$$

$K = 0.6363$
$D = 24$
$Y_A = 0.9999$
$F_{AA} = 1.008$ (from Appendix E)
$h_w = 15$
$v = 1.189$ ft³/lb (source: Steam Flex)

$$\begin{aligned}\dot{M} &= 5.982 \times 0.6363 \times 24^2 \times 0.9999 \times 1.008 \times \sqrt{\frac{15}{1.189}} \\ &= 7848.8\,\text{lb/min}\end{aligned}$$

This page intentionally left blank

Chapter 11

Troubleshooting

Table 11.1 is a guideline to help troubleshoot aerodynamic-related problems. As with any troubleshooting situation, there are some things that need to be done first.

1. *Define the problem.*
   a. *What exactly is the problem?*
   b. *What should the performance be?*
   c. *What is the performance now?*
2. *Outline the history of the compressor.*
   a. *How long has it been operating?*
   b. *When was the last overhaul?*
   c. *What changes were made at that time?*
   d. *When did the problem start?*
   e. *Was it a quick or gradual change?*
   f. *Note the trend of various parameters and how they relate to the original design conditions.*
   g. *Note how the head, efficiency, and work input deviate from design over time.*
   h. *What else changed, what other problems occurred at this time*
      i. *on the compressor?*
      ii. *in the process?*
      iii. *in operation and control?*
3. *Verify all data.*
   a. *Have instruments been calibrated?*
   b. *Do cross checks agree?*

If a thorough performance test has not been completed, do it now. Get help if it is needed. Be sure to follow as closely as possible the test procedure outlined in the previous chapter. If possible get several operating points at one speed so a full curve can be plotted. This can be a big help in determining corrective measures. Monitor the performance continuously over time to develop a detailed trend.

COMMON SOURCES OF TEST ERROR

In order to troubleshoot any problem, it is important to have correct information on the subject. Aerodynamic performance is very involved and data errors can rapidly mushroom, thereby misleading the problem solver. It is therefore essential that accurate data be obtained.

Before troubleshooting the compressor, troubleshoot the testing procedure. The best way to do this is a power balance. If it is not feasible to do a power balance, or if there is a significant error (7% or more) between the compressor power and the driver power, a thorough analysis of the test procedure is necessary. Plotting the compressor work (Eq. 7.10) may be helpful.

Gas Analysis

To have good test results, it is critical to have an accurate gas analysis. This can be a bit complex on high-pressure, high-mole-weight gas. If the sample is taken at high temperatures, some of the heavy gas may condense on the walls of the sample container when it cools. If you take the sample at the inlet, there may be some liquids in the gas stream that will remain in the sample container. When you test the gas, this condensed liquid will remain in the bottle unless heated.

Compressor Performance. https://doi.org/10.1016/B978-0-12-814219-6.00011-8

TABLE 11.1 Troubleshooting Checklist
• Define problem—what, where, when
• Outline history of operation—trend data
• Verify data
Test accuracy
Compare work input to predicted values
Complete power balance Check pressure taps: location, size, condition
Liquid in pressure lines
Temperature probe insertion depth, heat transfer
Calibrate instruments
Inspect flow meter: wear, sludge buildup
Condensates in gas analysis
Vortex or undeveloped velocity profile upstream of flow meter
Conduct a mass flow balance
Equipment problems
Internal leakage across diaphragm splitline
Recirculation from rubbed interstage seals or balance piston seals, casing drains, other areas
Foreign object damage or blockage
Liquids in process
Dirt accumulation or polymer buildup
Erosion of impeller blades and diffuser passages
Vortex or undeveloped velocity profile upstream of compressor suction
Proper direction of rotation
Balance line sleeve
Economics
Per diem cost to operate as is
Associated risks
Cost for repairs
Cost for down time to complete repairs
Safety concerns

For best results, take samples at both the inlet and discharge points. Check for condensibles and compensate by heating the sample before testing.

Liquids in the System

If there is liquid anywhere in the system, it is possible that some may carry over into the compressor. Knockout drums and demister pads do not always work the way they should. This liquid carryover will give erroneous results on the performance test. One way to be certain there is no liquid is to measure the flow rate at each flange, because liquids in the gas stream will also give an erroneous reading to the flow meters. This mass flow balance is a good crosscheck for both flow measurement accuracy and/or possible liquid ingestion. An instrument for checking for liquid is shown in Fig. 11.1.

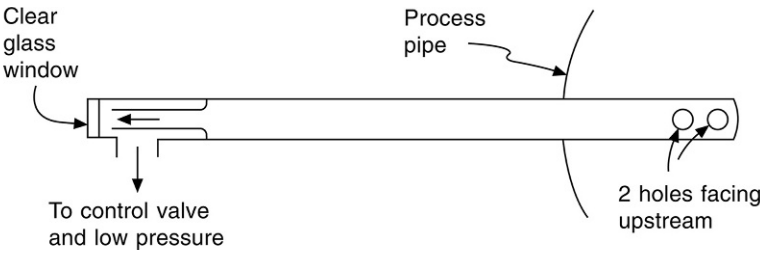


FIG. 11.1 Instrument for detecting liquids in process piping. Liquid will cause the viewing window to fog. *(Data from Lock JA. Techniques for more accurate centrifugal compressor performance evaluation. Houston, TX: Southwest Research, Turbomachinery Symposium; 1981.)*

Another liquid problem is liquid in pressure tap lines. Be sure all lines are properly sloped and drained. If lines are too small (less than 1/2 in.), capillary action will hold liquid in the lines.

Be sure to open drain valves at low spots in process piping and instrument lines before, during, and after test.

Pressure and Temperature Measurement

Be sure that a proper pressure tap is installed (see Fig. 7.2). Inspect the inside edge of the hole to see that it was deburred and that it has not been eroded, corroded, or plugged with dirt.

Check thermocouple installations. Thermocouples should be inserted into the pipe 1/3–1/2 of the pipe diameter. Also, it is critical that the thermocouple have intimate contact with the thermowell. Use graphite paste as suggested in Chapter 7.

Although not normally required, the proper method of analyzing compressor performance utilizes "total" pressure and temperature rather than the "static" values that are measured (see Eqs. 7.23, 7.24). Total values are usually only required when very high velocities are encountered.

Be sure to use only instruments that have recently been calibrated. Significant errors can be introduced by normal vibration and handling during operation.

As a reference, Fig. 11.2 is provided to demonstrate the effect of pressure and temperature error on performance test results.

Velocity Profile

If possible, a flow meter should be installed in each inlet and discharge pipe so a mass flow balance in the system can be carried out. This is done by simply comparing the total mass inflow to the total mass outflow. The difference is the accuracy of the flow meters.

A major source of problems both in compressor performance and in obtaining accurate flow data is an incomplete velocity profile and/or a vortex upstream of the compressor suction or flow meter. Either situation can seriously alter the compressor and/or flow meter performance.

A flow straightener device is required when flow swirl or a vortex is present. This can occur when there are two or more adjacent elbows in different planes. A flow straightener can be a tube bundle or an "egg crate," as shown in Fig. 11.3.

A flow equalizer is required when the velocity profile is not uniform. This can be caused by flow hugging one side of a pipe due to flow around an elbow or flow through a partly closed butterfly valve. This situation is best corrected by an equalization plate, which is essentially a perforated plate. Think of it as parallel orifices in a flow path. At higher velocities, the resistance (pressure drop) is greater. The higher velocity side of the velocity profile is restricted more than the lower velocity side, causing a shifting and equalization of the velocity profile (see Fig. 11.3).

When designing a flow equalizer, it is important to realize that pressure drop can be significant, especially if the plate becomes plugged with debris. The best method for calculating pressure drop is to add the area of all the holes in the plate and determine an equivalent single hole orifice while calculating pressure drop accordingly (see Chapter 10). Be sure to note the effect of the recovery factor (Fig. 10.6).

Make a schematic diagram of the compressor and adjacent piping. Note the length of straight runs of pipe, elbows, flow meters, valves, suction strainers, knockout drums, silencers, flow straighteners, instrumentation, etc. This will help in resolving system-related problems.

To ensure that a proper length of straight piping run is available upstream of the compressor, refer to Chapter 6, inlet piping. Table 11.2 is for orifice meters (reference ASME "Fluid Meters" for straight runs required upstream for an orifice with a 0.5 beta ratio). For minimum straight run of pipe required, see Table 11.2 and (Fig. 10.2 for specifics, or refer to ASME "Fluid Meters."

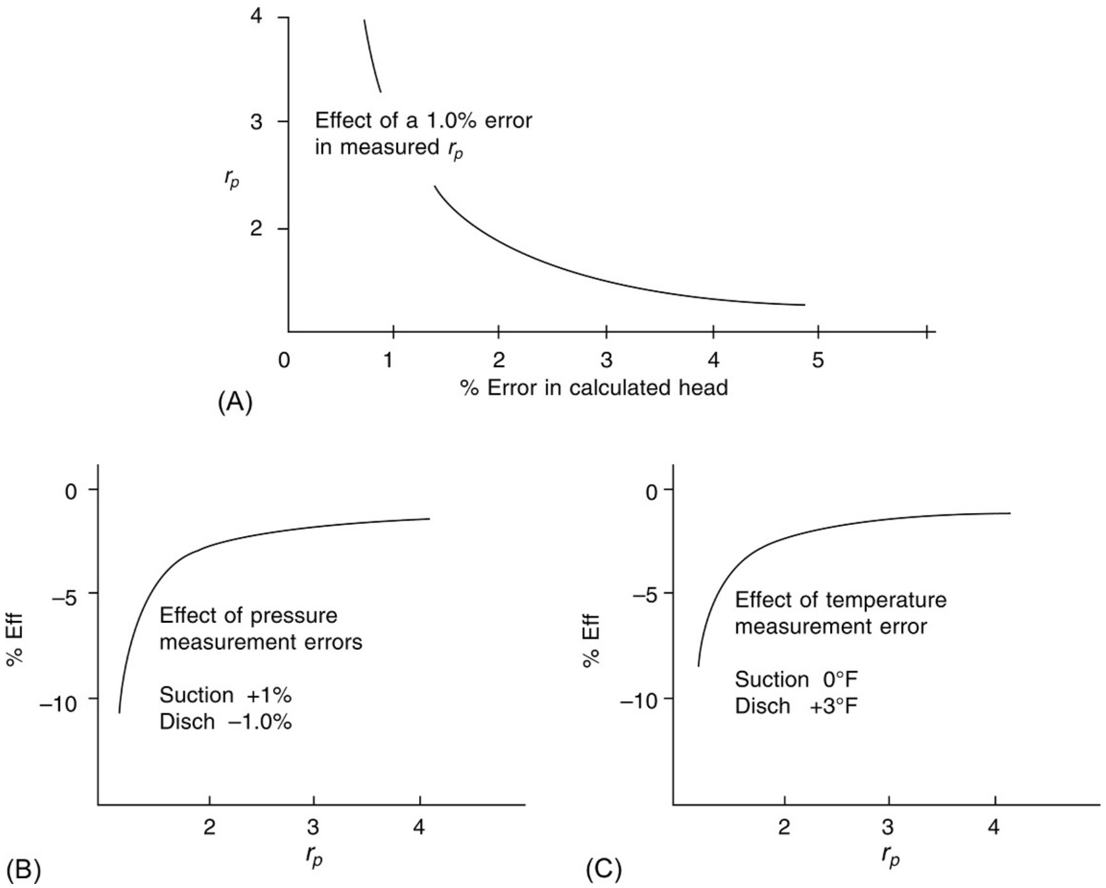


FIG. 11.2 Effects of measurement errors on test results. Data for natural gas (based on information from Lock [26]): (A) Effect of a 1% error in measured pressure/ratio on calculated isentropic head. (B) For the suction and discharge measurement errors shown, the percentage point deviation in efficiency is plotted versus pressure ratio. (C) Percentage point deviation in efficiency for the temperature measurement errors shown.

FIG. 11.3 Vortex and nonsymmetric velocity profile caused by piping arrangement.

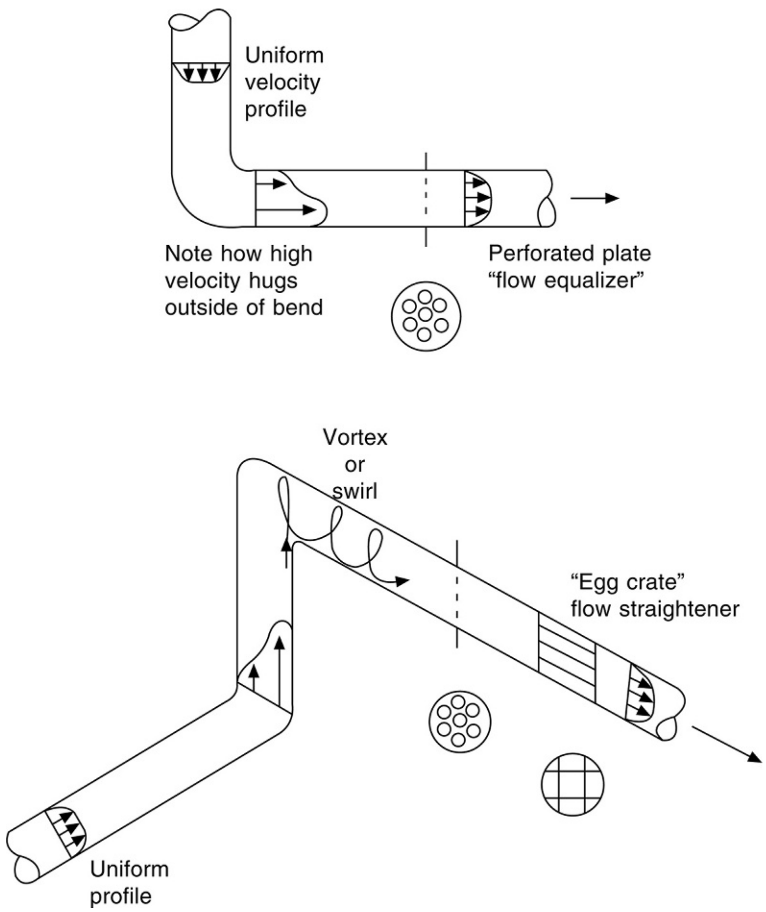


TABLE 11.2 Approximate Straight Run of Pipe Required Upstream of Orifice Plate

Piping Arrangement	Diameters of Straight Run
Long-radius elbow	7
Two elbows—same plane	8
Two elbows—different plane	12
Partially closed valve	25
Gate valve	6.5
Check valve	11

Check Mechanical Operating Data

Check the trend of the axial position of the rotor and compare to the trend of the compressor efficiency. Mutual changes in performance and thrust position may indicate a balance piston or interstage labyrinth seal rub. Does the balance line ΔP (flow) also follow a similar pattern? What about the thrust-bearing temperature? All of these items can indicate internal damage.

CLEANING AXIAL AND CENTRIFUGAL COMPRESSORS

Sometimes dirt, polymer buildup, or other substances can clog the compressor internally and seriously degrade performance. Very small amounts of dirt on axial blades alter the blade profile and degrade performance. Cleaning a compressor may be all that is required to regain "like new" performance.

Organic Abrasives

An axial or centrifugal compressor can be easily cleaned during normal operation (design speed) by using either uncooked rice or walnut shells as an abrasive agent. Depending on the process for which the compressor is used and the extent of contamination permitted to the process, the compressor can be cleaned with air going to the process or through the atmospheric blowoff valve. This must be determined by the user. Proceed as follows:

1. Locate a suitable piping or instrument connection at the inlet pipe after the filter where a funnel with a 1/2-inch **ID** spout or smaller can be introduced. This connection will be used to introduce the abrasive material and, therefore, would preferably be located at the top centerline of a horizontal piping run. This will allow the abrasive agent to fall into the air stream and prevent any accumulation of material at a point where it may be pulled into the machine as one large mass.

   If no suitable connection is available, one of the second stage filter elements may be removed to allow the introduction of material. When using this method, be certain that personnel entering the filter house have no loose objects or clothing, rags, pens, etc., which may be sucked into the compressor suction.

   If suction pressure is above atmospheric pressure, an ejector may be used to inject the material.
2. Position an operator where he can monitor the air flow and discharge temperature through the machine. Note the air flow and discharge temperature through the machine prior to introducing any foreign material. Vibrations should be monitored for any abnormality during the cleaning. If any vibration increases suddenly, suspend the cleaning immediately until the source of the vibration is determined and corrected.
3. Slowly begin introducing the walnut shells or uncooked rice through the funnel at a rate not to exceed *one pound per minute*. The largest dimension of the abrasive should be no greater than one-quarter inch, or one-half the size of the smallest passageway in the compressor, whichever is smaller.
4. Stop introducing the abrasive after five pounds and check the difference in air flow and discharge temperature through the machine (this assumes that speed, discharge pressure, and stator vane setting have remained constant). Note the variation in flow and temperature from the flow noted in item 2.

Repeat steps 2, 3, and 4 until no further upward change is noted in air flow through the machine and a decrease in discharge temperature is observed.

Depending on the internal cleanliness of the machine, the air flow rate may change rapidly with the introduction of the abrasive (walnut shells or uncooked rice). Therefore, if the machine is on-line to the process, care must be taken that a process upset does not occur due to large changes in air flow rates.

When introducing the abrasive into the air flow stream, be certain to introduce the material in a steady flow. It is important that the abrasive does not enter the air stream as a large mass, clump, or batch. This will cause damage to the blading.

Proper preparation by the persons introducing the material will avoid problems. It is suggested that a “dry run” be made prior to the actual cleaning to determine the funnel size best suited to the abrasive particle size used. This will prevent plugging of the funnel or “batch” introduction of abrasive material and help to establish a better understanding of exactly what is required.

Liquid Wash

For cleaning of centrifugal compressors, note that this procedure will only clean the gas passages. It will not clean between the impeller(s) and diaphragm (or backplate). Material buildup in this area can result in a rub and high vibration. In order to clean this area, a liquid wash or steam cleaning is required.

To liquid wash, fill the compressor with a suitable solvent (hot condensate) almost to the horizontal centerline, but below the seal level. A manometer can be used for this. Provide buffer to seals so liquid does not get into bearing and seal cavities.

While injecting the liquid solvent, rotate the shaft at 20 RPM or less. After 15 min, drain all liquid from casing. Sample the liquid drained from the casing for suspended and dissolved particles. Repeat the process until the sample shows suspended and dissolved particles are at a minimum.

For single-stage compressors, it is most effective to inject liquid solvent or steam via a port between the backplate and impeller. Liquid through spray nozzles or steam may be injected to this area on a continuous basis to minimize buildup in this area. Be cautious, however, of erosion effects.

For cracked gas compressors, coke oven gas compressors, and other applications where buildup on the diffuser surface is a problem, continuous flushing with solvent may be required. For continuous duty, a maximum solvent injection should be no more than 3% of compressor mass flow. Steam has been successfully used as a solvent for some situations such as coke oven gas. To minimize polymer buildup, even water has been used. This may help by providing a “wet” surface and thereby dissuading adherence of the polymer. The results of evaporative cooling may also be a factor. Coating diaphragm surfaces with an antistick compound may provide further resistance to fouling.

For severe fouling, the compressor must be disassembled and cleaned by mechanical means and/or by a high-pressure spray gun utilizing solvents, caustic, steam, or just plain water. Be sure to pull the diaphragm and clean the return channels and all other flow areas. To ease future disassembly, place plastic or aluminum food wrap between diaphragms and casing. Be sure to check temperature/chemical compatibility.

Caution: These abrasive cleaning procedures are not recommended for small, high-speed compressors. The high-speed, light weight, and close clearances make this type of compressor very sensitive to foreign object ingestion. Organic abrasive cleaning of such a compressor could result in equipment failure. This type of compressor should only be cleaned after disassembly. There is no need to remove the rotor. Simply remove the inlet casing and clean the impeller and diffuser areas. Clean with a mild abrasive detergent and flush clean. Reassemble.

Always check with the equipment manufacturer first before any abrasive or liquid cleaning method is attempted.

INSPECTION OF COMPRESSOR

Once you are certain that you have good data, the unit is clean, and the axial position and thrust-bearing temperatures have been checked, it will be necessary to open it up and look inside.

Visual Inspection

The following inspection points are to be completed while the top half of the casing is being lifted or immediately after and before any cleaning or further disassembly is done. No special tools or templates are required, as these are visual observations only.

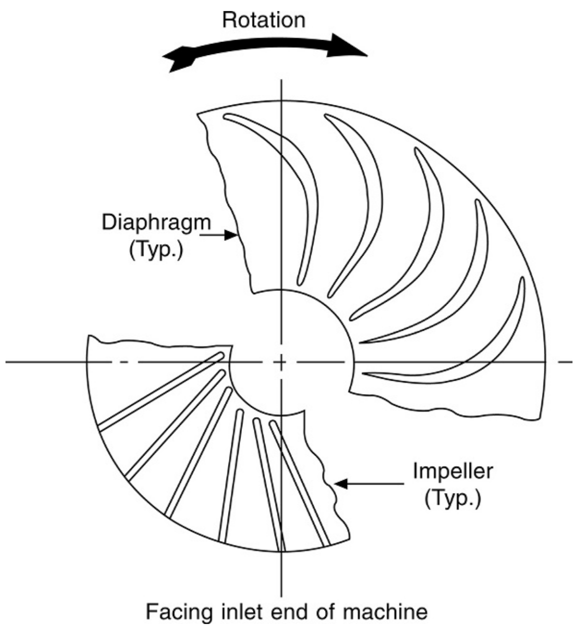


FIG. 11.4 Check for proper rotation. *(Reprinted with permission of Elliott Company, Jeannette, PA.)*

Check for proper rotation of impeller and return channel vanes. Diffuser vanes should also be checked if they are present. Note that impeller vanes are usually backward-leaning. Right- and left-side diaphragms of double-flow compressors can be (and have been) interchanged (Fig. 11.4).

Check for blockages in the flow passages. Many things have been found in compressors: T-shirts, lunch boxes, valve parts, gloves, etc.

Check to see that recirculation is not possible through the casing drain system. On barrel compressors, this includes interstage leakage paths in the area between inner and outer casings.

Check the general cleanliness of the internals. Note the location and extent of rust, corrosion, and polymer buildup. If possible, take samples of the polymer, as this can help in determining the proper solvent to be used.

Note locations of erosion, which may indicate that liquid or particles are being ingested. Erosion of impeller blades and diffuser passages can seriously affect performance by a change in dimension or change in surface finish.

Check for visual signs of recirculation across the diaphragm splitline. Streak marks can often be seen if there is flow across the splitline.

Some compressors, such as the Elliott horizontally split models, depend solely on the weight of the upper-half diaphragm to provide a seal at the diaphragm horizontal splitline. As the top-half casing is being lifted, check to see that the top-half diaphragms are free to fall below the horizontal splitline as far as is allowed by the antirotation hardware. If the diaphragms seem to be locked in place, they should be removed and cleaned so that free movement is allowed. Note the use of sealant or O-rings at the diaphragm splitlines.

Note general condition of shaft, impeller eye, and balance piston seals. Note condition of any O-rings or sealing strips that isolate the balance piston chamber area from the volute.

Check for anything that may have kept the splitline apart (dirt, tools, hardware, etc.). If sealant was used on the splitline, check for sections missing. These conditions can provide a recirculation path and degrade performance.

One area that can create problems is the balance line sleeve used in some compressors. Omission of this sleeve or O-ring during assembly can result in excessive balance piston leakage, seal oil leakage, and thrust loads (Fig. 11.5).

Dimensional Inspection

Record shaft, impeller eye, or blade tip, and balance piston seal clearances. Top and bottom can be done by lead wire, and right- and left-side clearances by feeler gauge. Maximum and minimum clearances can be found in the operating manual.

For open-wheel impellers and axial compressors, blade clearance to stationary hardware can be critical. Check clearance values according to factory specifications. The approximate effect on efficiency is shown in Fig. 11.6 and the following equations [27,28].

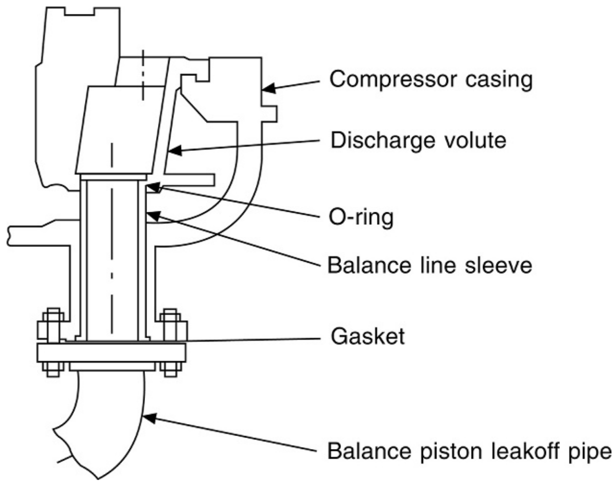


FIG. 11.5 Balance piston line sleeve found in some centrifugal compressors [18]. *(Reprinted with permission of Elliott Company, Jeannette, PA.)*

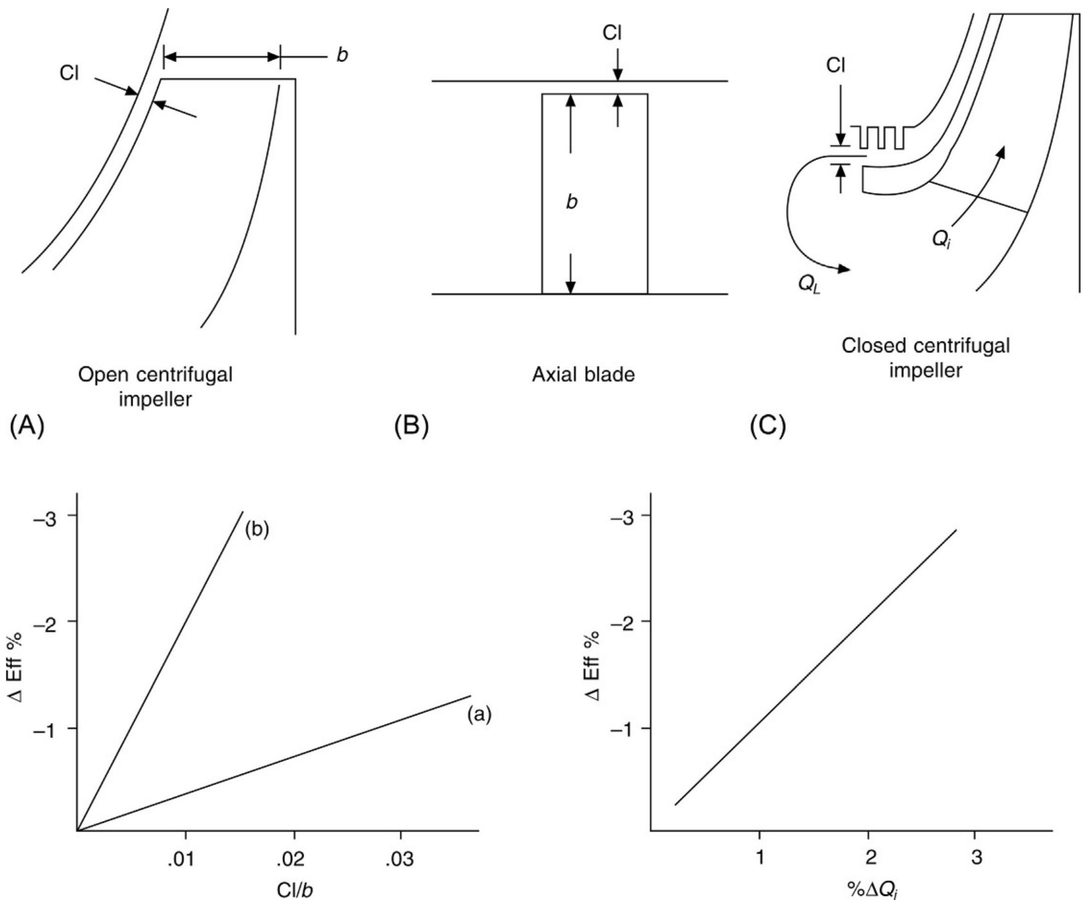


FIG. 11.6 Effect of blade tip clearance. (A) For an open centrifugal impeller, the efficiency loss is only one-third of a point for each percent of tip clearance ratio at the impeller outer diameter. (B) For an axial compressor blade, the efficiency loss is about two percentage points for each percent of tip clearance ratio. This is also true for the throat region of an open centrifugal impeller. (C) For a closed centrifugal impeller, the efficiency loss is about one percentage point for each percent increase in Q_i (as a result of Q_L increasing and inlet flow to the compressor constant). *(Data from Ishida M (Nagasaki University), Senoo Y (Kyushu University). The pressure losses due to the tip clearance of centrifugal blowers. New York: American Society of Mechanical Engineers; 1980. Lakshminarayana B (Pennsylvania State University). Methods of predicting the tip clearance effects in axial flow turbomachinery. New York: Journal of Basic Engineering, American Society of Mechanical Engineers; 1970.)*

For Axial compressors and centrifugal compressors at the impeller eye (open wheels), the compressor efficiency is reduced by 2 percentage points for each percent of tip clearance ratio.

$$\%\Delta\eta = -2 \times (\mathrm{Cl}/b)\%$$

where

$$Cl = \text{Blade tip clearance, in.}$$

$$b = \text{Blade height, in.}$$

$$(\mathrm{Cl}/b)\% = \frac{\mathrm{Cl}}{b} \times 100\%$$

The tip clearance for centrifugal impellers is less critical. The efficiency reduction is only one-third of a point for each percent of tip clearance ratio.

$$\%\Delta\eta = -\frac{1}{3} \times \frac{\mathrm{Cl}}{b}\%$$

For closed centrifugal impellers with labyrinth seals at the eye, the following rule of thumb may be used:

$$\%\Delta\eta \propto -\%\,\Delta Q_i$$

where
$\%\Delta Q_L \propto \%\Delta \mathrm{Cl}_L$ (for small ΔCl)
Q_L = Labyrinth seal flow
Cl = Labyrinth clearance, radial
Q_i = Impeller flow

This states that for every percent increase in impeller through-flow due to increased labyrinth leakage, the efficiency will be reduced by about 1%.

Check the axial alignment of the impellers to the diffuser walls. Tolerances and reference points (which may vary from stage to stage) are shown on the assembly drawing. The thrust bearing and housing must be in place and the endwall bolts drawn up tightly when making this check.

Pull the flow meter(s). Check for buildup or corrosion. Measure the bore. Does it check to specifications? Does the square-edged orifice have a square edge or is it rounded off from erosion? Inspect the pressure taps. Are they clean and in good condition? Inspect the pipe upstream and downstream of the flow meter. Is there any debris, dirt, or sludge buildup in the pipe? This can seriously affect performance of the flow meter.

ECONOMICS

The primary responsibility of each employee is to help keep the plant operating efficiently. That means material and labor going in one end of the plant and money coming out from the other. The plant manager has a hard time justifying anything that "looks good" or "might make the equipment run better." Project requests should be in terms of money.

- How much does it cost to operate "as is" versus the cost of operating the equipment after the proposed project is completed?
- What are projected expenses of problems that might occur if the project is not completed?
- What are costs for repairs—parts, labor, time?
- What are costs for down-time for the project?
- Are there safety concerns if the project is not completed?
- What kind of "insurance" is available? This could be in the form of spare parts, special technical help, maybe a spare machine.

Remember to check all angles and put things in terms of cash. The right dollar amount can convince the plant manager of almost anything, or it may reveal that the project is not quite as important as was originally thought.

FIELD PROBLEMS

The following is a brief summary of several actual field troubleshooting experiences (Figs. 11.7–11.16 and Table 11.3).

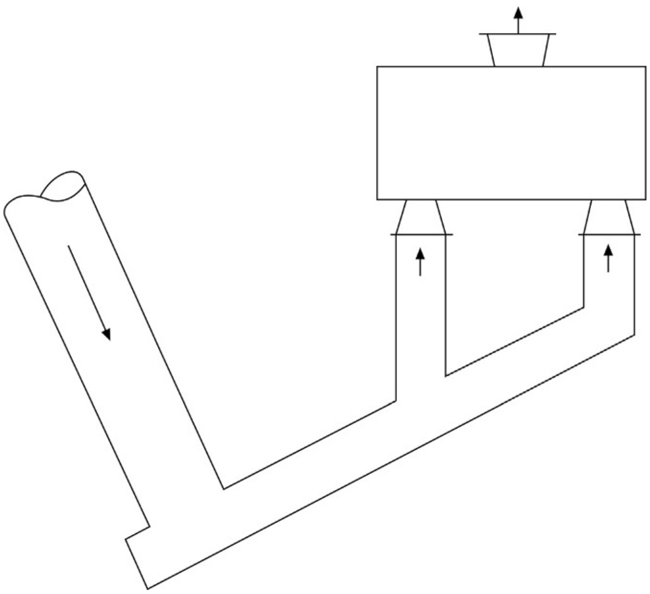


FIG. 11.7 Case study A—double-flow compressor inlet piping. *Problem*: During commissioning, a double-flow compressor was found to be low in head. Also, the unit surged prematurely. *Discussion*: The inlet piping caused unequal flow distribution to the compressor inlets. This resulted in one section's running near surge while the other was operating near the overload region. *Resolution*: The inlet piping to the compressor was modified to improve flow distribution.

FIG. 11.8 Case study B—duplicate compressors—mirrored inlet piping. *Problem*: Two duplicate single-stage air compressors were found to have a significant capacity difference during commissioning. *Discussion*: Both compressors had been performance tested at the factory and were within 1.0% of each other. The suction piping for each unit was identical to the other except that they were mirror images. The axial inlet compressors had two elbows at different planes and a suction throttle valve. This piping arrangement caused flow swirl, which caused prewhirl at the impeller and affected the head output. *Resolution*: The inlet piping was modified to include mitered elbows, which minimized the problem [11].

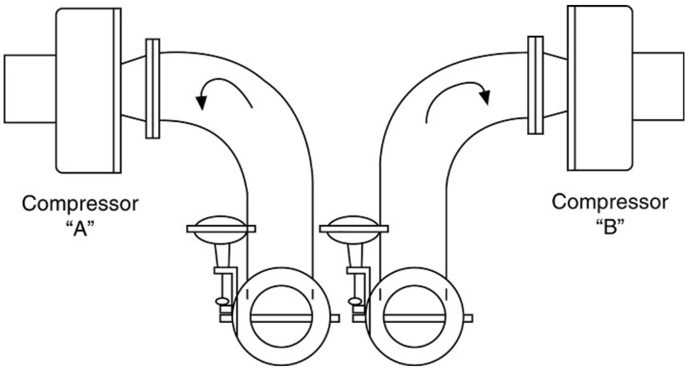


FIG. 11.9 Case study C—vortex separator. *Problem*: During commissioning, a high-pressure multistage compressor was found to be low in head and efficiency. Near surge, control became unstable. The compressor would rumble and continue to surge even with the recycle valve open. Eventually, it would trip on high vibration. *Discussion*: It was found that a vortex separator was being used upstream of the compressor to assure liquids did not enter the compressor. The residue vortex affected both the orifice and compressor performance. *Resolution*: Flow straightening vanes (egg crates) were utilized downstream of the separator to reduce the vortex.

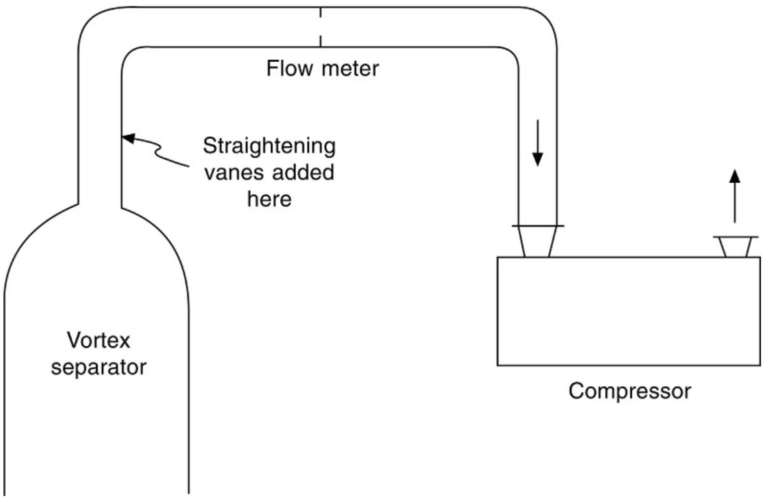


TABLE 11.3 Case Study D

	Before	After
Discharge pressure (PSIG)	410	410
Discharge temperature (°F)	142	116
Axial position (Mils)	24	19
Balance line ΔP (PSID)	4.7	1.5
Speed (RPM)	11,440	10,770
Thrust metal temperature (°F)	240+	165

Problem: Higher-than-normal speeds were necessary to maintain the required discharge pressure for a particular multistage compressor.
Discussion: Other data were reviewed. The unit also had a high thrust-bearing temperature reading. The axial position of the shaft was abnormally high, as well as the balance line ΔP. The balance piston or interstage labyrinth seals were suspected to be oversized due to a bad rub.
Resolution: The next turnaround found the impeller seals and balance piston seals damaged. Long-term solution was to install abradable seals.

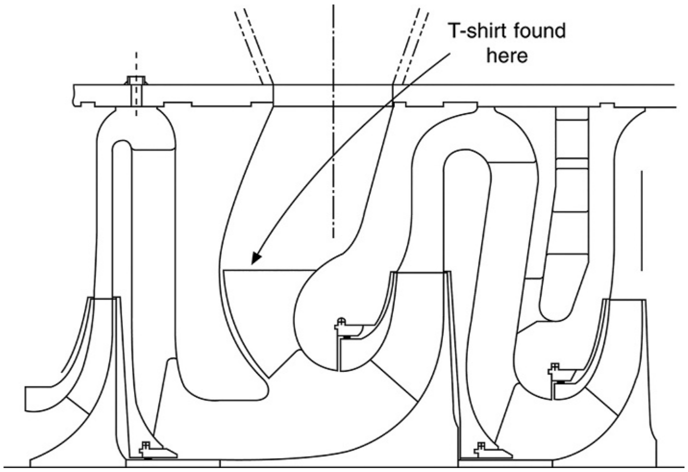


FIG. 11.10 Case study E—inlet blockage. *Problem*: A multisection multistage centrifugal compressor had low head on one section. *Discussion*: Since the unit had just been rerated, a design or manufacturing problem was suspected. The compressor was disassembled and inspected. *Resolution*: A worker's T-shirt had been found partially blocking the inlet guide vanes on the section, which was low in head.

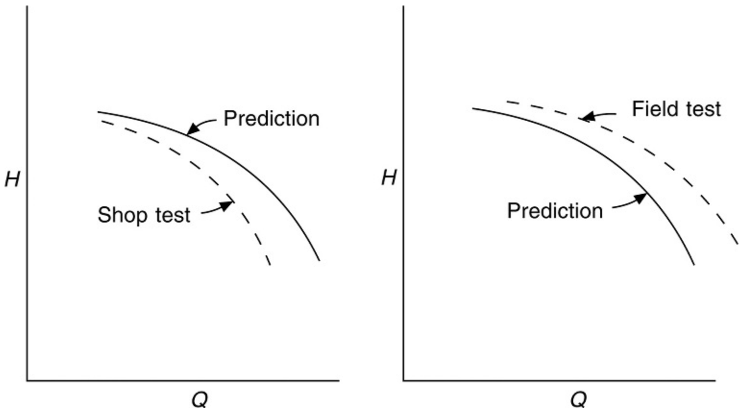


FIG. 11.11 Case study F—factory test. *Problem*: Upon commissioning, a multistage compressor was found to be high in head and capacity, causing the motor driver to overheat. *Discussion*: During the factory performance test, the compressor had been found to be low in head and capacity. Adjustments were made to meet the required guarantee point. Quite some time later, it was discovered that an error had been made in the required piping upstream of the flow orifice for the shop performance test. This piping caused flow swirl and corresponding erroneous flow measurements during the shop testing. *Resolution*: The compressor power was reduced by changing speeds via a gear change.

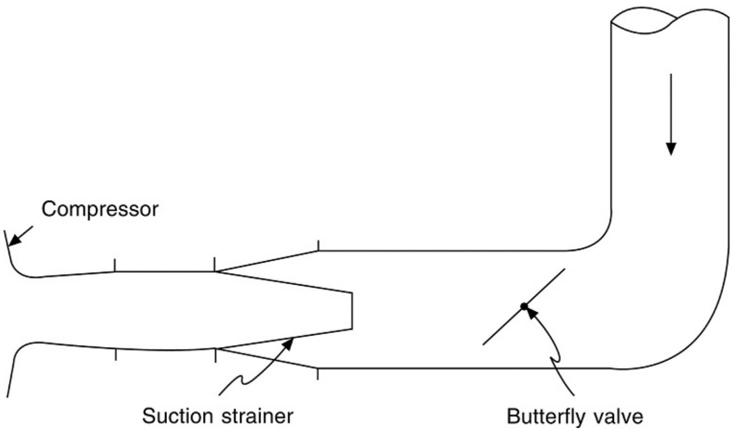


FIG. 11.12 Case study G—suction strainer. *Problem*: A single-stage compressor was found to be significantly low in head. *Discussion*: In addition to aerodynamic problems, the compressor had a serious mechanical problem as well. The impeller was found to be developing cracks. The inlet piping had a strainer and a butterfly valve near the suction. Upstream of this was an elbow. *Resolution*: The inlet piping was modified by replacing the suction strainer with an equalizing plate. Also, the butterfly valve was removed and put further upstream.

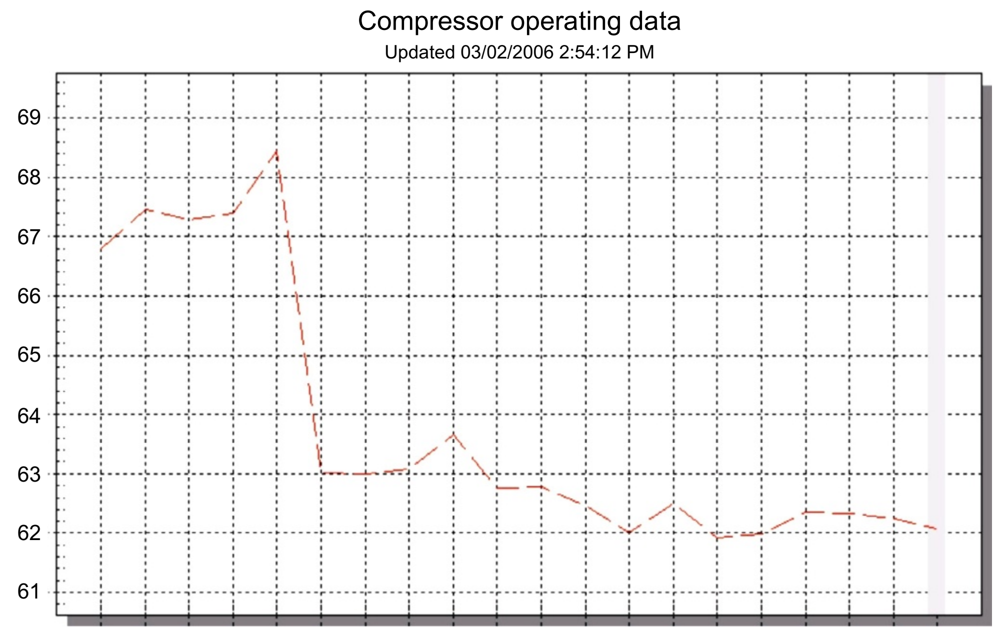


FIG. 11.13 Case study H—rusty pipes. *Problem*: The plant capacity was found to have suddenly dropped off overnight. *Discussion*: The main gas compressor was suspect and since the compressor performance was continuously monitored with Gas Flex, the performance data were quickly reviewed and provided the information needed to justify immediately going into the compressor. Rust and sludge was found in the compressor that apparently came from corrosion in the inlet piping. *Resolution*: The compressor was opened, cleaned and the spare rotor installed. The inlet piping was cleaned and insulated to prevent further water condensation and corrosion.

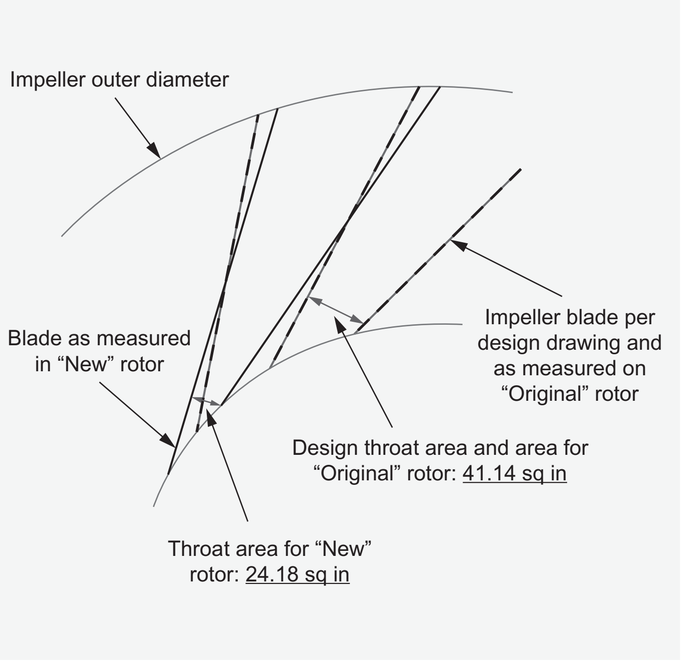


FIG. 11.14 Case study I—impeller fabrication error. *Problem*: Following turnaround a new, never-been-run spare rotor was installed. Capacity was restricted to extremely low flows, below the normal surge point. *Discussion*: The system piping and controls were thoroughly checked and eventually the rotor was pulled and taken to the OEM shop. CMM measurements on the entire rotor revealed a restriction in the impeller throat. *Resolution*: The original rotor was installed in the compressor and a new impeller was fabricated for the spare rotor.

FIG. 11.15 Case study J—insipient surge. *Problem*: Not long after commissioning, the main air blower suddenly failed without warning. *Discussion*: During commissioning, the antisurge system was set up from the control room. No one was standing near the compressor to hear the compressor surge and it was not possible to define a distinct surge point from the control room using instrumentation alone. The compressor ran in insipient surge (flow separation on the leading edge of the blade) and eventually the blade failed causing failure of the compressor. *Resolution*: The impeller was redesigned with improved surge characteristics, reduced stresses in the blade root and dampening tubes on the blades.

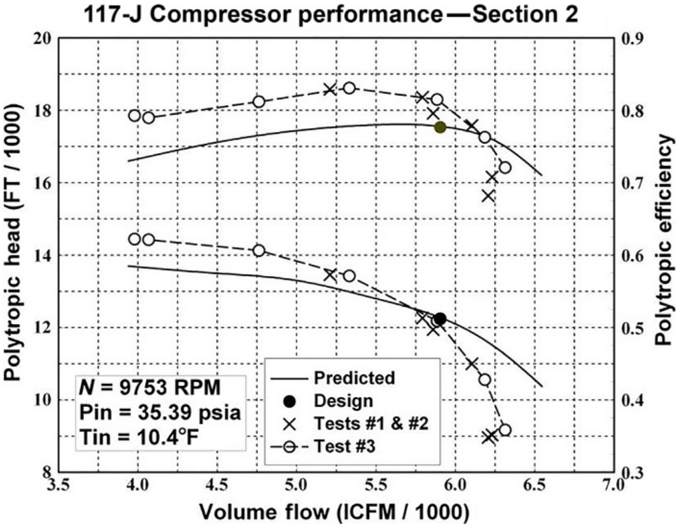


FIG. 11.16 Case study K—premature choke. *Problem*: During a shop performance test, the compressor was found to choke prematurely. *Discussion*: Analysis by the OEM revealed that area schedules on stationary hardware in the compressor were undersized. *Resolution*: Even though the compressor contractually met the design point guarantee, the compressor was disassembled, restricted areas were increased to allow the compressor to operate further out in the choke region as originally designed, and the compressor successfully retested.

MAINTENANCE CHECKLIST

Check for the following items:

- Preshutdown performance
- Interstage labyrinth seal clearance
- Balance piston seal clearance
- Axial blade tip clearance
- Blade tip clearance on open centrifugal impellers
- Internal splitline leakage
- Impeller to diffuser alignment
- Cleanliness of internal parts
- Surface finish of internal parts (pitting, corrosion, buildup of foreign material, etc.)
- Proper installation of stationary guide vanes, splitter vanes, etc.
- Performance at start-up

Troubleshooting Guide

Symptom	Possible Cause
A. Low head	1. Low suction pressure[a]
	2. High suction temperature[a]
	3. Mole weight below design[a]
	4. Internal recirculation[b]
	a. Damaged balance piston seal
	b. Damaged interstage seals
	c. Leakage across diaphragm splitline
	5. Design deficiency
	6. Error in test data
	7. Flow swirl at suction[c]
	8. Internal blockage or corrosion[b]
	9. Nonuniform flow profile at suction[c]
	10. Error in flow data
	11. Impeller (or rotor blades) damaged, corroded, or dirty[c]
	12. Diffuser (or stator vanes) damaged, corroded, or dirty[b]
	13. Damaged or incorrect inlet or return channel vanes
B. Low capacity	1. See A above
C. High power (driver and compressor)	1. High suction pressure
	2. Low suction temperature
	3. High mole weight
	4. Internal recirculation (see A4)
	5. Internal blockage, corrosion, or dirt buildup
	6. Liquid ingestion
D. Low efficiency	1. See A
E. Poor power balance	1. High gear losses
	2. Incorrect gas analysis
	3. Incorrect flow data
	4. Inaccurate gas properties or calculation procedure
	5. Incorrect driver power
F. Premature surge	1. Increased suction pressure[d]
	2. Reduced suction temperature[d]
	3. Increased MW[d]
G. Premature choke	1. See A
H. Reduced operating range	1. See A
	2. See F
I. Low power	1. Impeller blades or axial rotor blades damaged, corroded, or dirty
	2. Flow swirl at suction
	3. Low suction density
	4. Error in test data
	5. Nonuniform flow profile at suction

[a] *See Fig. 4.16.*
[b] *If work input/power is per predicted values, and head and efficiency are low, the problem is probably recirculation, blockage, or corrosion.*
[c] *If work input/power is low, the problem is centered around something that affects the ability of the impeller/rotor blades to do work—flow swirl, damaged or incorrect impeller/rotor blades, etc.*
[d] *See Fig. 4.18.*

Part III

Reference Material

This part of the book is provided to aid in the determination of gas properties. For best results, it is suggested that one of the many computer programs available for personal computers, such as Gas Flex, be used. The data provided here, however, can produce acceptable results for many cases.

This page intentionally left blank

Appendix A

Gas Properties

Three separate areas are covered in Appendices A-C.

(Tables A.1A, A.1B, A.2–A.4 and Fig. A.1.) Data required for the calculation of gas properties for pure gas or gas mixtures are provided (refer Chapters 2 and 7).

For reference, the specific volume for various gases is listed. Also, the viscosity is tabulated for selected gases at various conditions.

THERMODYNAMIC STATE EQUATIONS

Thermodynamic state equations are developed from experimental data or derived from kinetic theory or statistical mechanics. The thermal equation of state relates state variables, usually, temperature, T, pressure, P, and density, ρ. The caloric equation of state relates the energy content of the fluid to state variables, usually in terms of specific internal energy, e, or specific enthalpy, h [35].

Thermally Perfect Gas

The most commonly used thermal equation of state is the perfect gas equation, $P = \rho RT$ (Eq. 2.5). Its limitation for real fluids is best realized by the fact that all gases can be liquefied. The highest temperature at which liquid and vapor can coexist defines the critical point for the fluid. From observed critical point properties (T_c, P_c, ρ_c), it is known that all gases are far from thermally perfect at this point. Use of the thermally perfect gas model should be limited to temperatures much higher than T_c and pressures much less than p_c to ensure reasonable accuracy. Basic thermodynamics shows that the energy content of a thermally perfect fluid is a function of temperature only, i.e., $h^0 = h^0(T)$. (For real gas models, the superscript, 0, is used to designate parameters for thermally perfect gases) [35].

Real Gases

The general form of the thermal equation of state for real gases is $Pv = ZRT$ (Eq. 2.7), where the gas compressibility factor, Z, may be obtained from established data (Fig. A.1) or from an appropriate real gas equation of state. A highly accurate real gas model is the eight-parameter Benedict-Webb-Rubn (BWR) equation

$$P = \rho RT + \left(B_0RT - A_0 - C_0/T^2\right)\rho^2 + (bRT - a)\rho^3 + a\alpha\rho^6 + \frac{c\rho^3}{T^2}\left(1 + \gamma\rho^2\right)e^{-\gamma\rho^2}$$

where A_0, B_0, C_0, a, b, c, α, γ are constants for the gas or gas mixture.

For real gases, h is a function of pressure as well as temperature. The effects of pressure are expressed as differences with respect to the perfect gas state (h^0) and referred to as departure functions [35].

TABLE A.1A Tabulation of Gas Properties in English Units

Gas or Vapor	Hydrocarbon Reference Symbols	Chemical Formula	Molecular Mass	Specific Heat Ratio $k = c_p/c_v$ at 60°F	Critical Conditions		[a]$C_{p,m}$	
					Absolute Pressure p_c (psia)	Absolute Temperature T_c (°R)	At 50°F	At 300°F
Acetylene	C_{2-}	C_2H_2	26.05	1.24	905	557	10.22	12.21
Air		$N_2 + O_2$	28.97	1.40	547	239	6.95	7.04
Ammonia		NH_3	17.03	1.31	1636	731	8.36	9.45
Argon		A	39.94	1.66	705	272	4.97	4.97
Benzene		C_6H_6	78.11	1.12	714	1013	18.43	28.17
iso-Butane	iC_4	C_4H_{10}	58.12	1.10	529	735	22.10	31.11
n-Butane	nC_4	C_4H_{10}	58.12	1.09	551	766	22.83	31.09
iso-Butylene	iC_{4-}	C_4H_8	56.10	1.10	580	753	20.44	27.61
Butylene	nC_{4-}	C_4H_8	56.10	1.11	583	756	20.45	27.64
Carbon dioxide		CO_2	44.01	1.30	1073	548	8.71	10.05
Carbon monoxide		CO	28.01	1.40	510	242	6.96	7.03
Carbureted water gas (1)		–	19.48	1.35	454	235	7.60	8.33
Chlorine		Cl_2	70.91	1.36	1119	751	8.44	8.52
Coke oven gas (1)		–	10.71	1.35	407	197	7.69	8.44
n-Decane	nC_{10}	$C_{10}H_{22}$	142.28	1.03	320	1115	53.67	74.27
Ethane	C_2	C_2H_6	30.07	1.19	708	550	12.13	16.33
Ethyl alcohol		C_2H_5OH	46.07	1.13	927	930	17	21
Ethyl chloride		C_2H_4Cl	64.52	1.19	764	829	14.5	18
Ethylene	C_{2-}	C_2H_4	28.05	1.24	742	510	10.02	13.41
Flue gas (1)			30.00	1.38	563	264	7.23	7.50
Helium		He	4.00	1.66	33	9	4.97	4.97
n-Heptane	nC_7	C_7H_{16}	100.20	1.05	397	973	39.52	53.31
n-Hexane	nC_6	C_6H_{14}	86.17	1.06	440	915	33.87	45.88
Hydrogen		H_2	2.02	1.41	188	60	6.86	6.98
Hydrogen sulfide		H_2S	34.08	1.32	1306	673	8.09	8.54
Methane	C_1	CH_4	16.04	1.31	673	344	8.38	10.25
Methyl alcohol		CH_3OH	32.04	1.20	1157	924	10.5	14.7
Methyl chloride		CH_3Cl	50.49	1.20	968	750	11.0	12.4

TABLE A.1A Tabulation of Gas Properties in English Units—cont'd

Gas or Vapor	Hydrocarbon Reference Symbols	Chemical Formula	Molecular Mass	Specific Heat Ratio $k = c_p/c_v$ at 60°F	Critical Conditions: Absolute Pressure p_c (psia)	Critical Conditions: Absolute Temperature T_c (°R)	$C_{p,m}$ At 50°F	$C_{p,m}$ At 300°F
Natural gas (1)		–	18.82	1.27	675	379	8.40	10.02
Nitrogen		N_2	28.02	1.40	492	228	6.96	7.03
n-Nonane	nC_9	C_9H_{20}	128.25	1.04	345	1073	48.44	67.04
iso-Pentane	iC_5	C_5H_{12}	72.15	1.08	483	830	27.59	38.70
n-Pentane	nC_5	C_5H_{12}	72.15	1.07	489	847	28.27	38.47
Pentylene	C_{5-}	C_5H_{10}	70.13	1.08	586	854	25.08	34.46
n-Octane	nC_8	C_8H_{18}	114.22	1.05	362	1025	43.3	59.90
Oxygen		O_2	32.00	1.40	730	278	6.99	7.24
Propane	C_3	C_3H_8	44.09	1.13	617	666	16.82	23.57
Propylene	C_{3-}	C_3H_6	42.08	1.15	668	658	14.75	19.91
Blast furnace gas (1)		–	29.6	1.39	–	–	7.18	7.40
Cat cracker gas (1)		–	28.83	1.20	674	515	11.3	15.00
Sulfur dioxide		SO_2	64.06	1.24	1142	775	9.14	9.79
Water vapor		H_2O	18.02	1.33	3208	1166	7.98	8.23

(Most values taken from Natural Gas Processors Suppliers Association Engineering Data Book-1972, Ninth Edition.)
(1) Approximate values based on average composition.
[a] *Use straight-line interpolation or extrapolation to approximate* $C_{p,m}$ *(in Btu/mol-°R) at actual inlet* T. *(For greater accuracy,* average T *should be used.)*
Adapted from Elliott multistage centrifugal compressors, Bulletin P-25C. Jeannette, PA: Elliott Co.; 1985.

The specific heat is calculated by using curve fit coefficients.

$$c_p = A + BT + CT^2 + DT^3 + ET^4 + FT^5$$

where A, B, C, D, E, and F are constants for the gas or gas mixture.

By combining the coefficients based on the gas mixture, gas mixture coefficients are established and used in the various equations to determine the gas mixture properties.

k Values

For real gases, there are three different k values. The three k values are equal for ideal gases; however, the values will diverge for real gases.

1. The first k value is the ratio of specific heats (c_p/c_v). This value varies only as a function of temperature. There is no pressure correction.
2. The second k value is the isentropic temperature exponent. k is calculated by the expression

$$\left(\frac{P_1}{P_2}\right)^{\left(\frac{k-1}{k}\right)} \frac{T_1}{T_2}$$

This value is normally used for flow element calculations. N method compressor head calculations also use this k value.

TABLE A.1B Tabulation of Gas Properties in Metric Units

Gas or Vapor	Hydrocarbon Reference Symbols	Chemical Formula	Molecular Mass	Specific Heat Ratio $k = c_p/c_v$ at 15.5°C	Critical Conditions: Absolute Pressure p_c (bar)	Critical Conditions: Absolute Temperature T_c (k)	[a]M_{cp} At 0°C	[a]M_{cp} At 100°C
Acetylene	C_{2-}	C_2H_2	26.05	1.24	62.4	309.4	42.16	48.16
Air		$N_2 + O_2$	28.97	1.40	37.7	132.8	29.05	29.32
Ammonia		NH_3	17.03	1.31	112.8	406.1	34.65	37.93
Argon		A	39.94	1.66	48.6	151.1	20.79	20.79
Benzene		C_6H_6	78.11	1.12	49.2	562.8	74.18	103.52
iso-Butane	iC_4	C_4H_{10}	58.12	1.10	36.5	408.3	89.75	116.89
n-Butane	nC_4	C_4H_{10}	58.12	1.09	38.0	425.6	93.03	117.92
iso-Butylene	iC_{4-}	C_4H_8	56.10	1.10	40.0	418.3	83.36	104.96
Butylene	nC_{4-}	C_4H_8	56.10	1.11	40.2	420.0	83.40	105.06
Carbon dioxide		CO_2	44.01	1.30	74.0	304.4	36.04	40.08
Carbon monoxide		CO	28.01	1.40	35.2	134.4	29.10	29.31
Carbureted water gas (1)		–	19.48	1.35	31.3	130.6	31.58	33.78
Chlorine		Cl_2	70.91	1.36	77.2	417.2	35.29	35.53
Coke oven gas (1)		–	10.71	1.35	28.1	109.4	31.95	34.21
n-Decane	nC_{10}	$C_{10}H_{22}$	142.28	1.03	22.1	619.4	218.35	280.41
Ethane	C_2	C_2H_6	30.07	1.19	48.8	305.6	49.49	62.14
Ethyl alcohol		C_2H_5OH	46.07	1.13	63.9	516.7	69.92	81.97
Ethyl chloride		C_2H_4Cl	64.52	1.19	52.7	460.6	59.61	70.16
Ethylene	C_{2-}	C_2H_4	28.05	1.24	51.2	283.3	40.90	51.11
Flue gas (1)			30.00	1.38	38.8	146.7	30.17	30.98
Helium		He	4.00	1.66	2.3	5.0	20.79	20.79
n-Heptane	nC_7	C_7H_{16}	100.20	1.05	27.4	540.6	161.20	202.74
n-Hexane	nC_6	C_6H_{14}	86.17	1.06	30.3	508.3	138.09	174.27
Hydrogen		H_2	2.02	1.41	13.0	33.3	28.67	29.03
Hydrogen sulfide		H_2S	34.08	1.32	90.0	373.9	33.71	35.07
Methane	C_1	CH_4	16.04	1.31	46.4	191.1	34.50	40.13
Methyl alcohol		CH_3OH	32.04	1.20	79.8	513.3	42.67	55.32
Methyl chloride		CH_3Cl	50.49	1.20	66.7	416.7	45.60	49.82

TABLE A.1B Tabulation of Gas Properties in Metric Units—cont'd

Gas or Vapor	Hydrocarbon Reference Symbols	Chemical Formula	Molecular Mass	Specific Heat Ratio $k = c_p/c_v$ at 15.5°C	Critical Conditions: Absolute Pressure p_c (bar)	Critical Conditions: Absolute Temperature T_c (k)	M_{cp} At 0°C	M_{cp} At 100°C
Natural gas (1)		–	18.82	1.27	46.5	210.6	34.66	39.54
Nitrogen		N_2	28.02	1.40	33.9	126.7	29.10	29.31
n-Nonane	*n*C_9	C_9H_{20}	128.25	1.04	23.8	596.1	197.07	253.10
iso-Pentane	*i*C_5	C_5H_{12}	72.15	1.08	33.3	461.1	112.09	145.56
n-Pentane	*n*C_5	C_5H_{12}	72.15	1.07	33.7	470.6	115.21	145.94
Pentylene	C_{5-}	C_5H_{10}	70.13	1.08	40.4	474.4	102.11	130.37
n-Octane	*n*C_8	C_8H_{18}	114.22	1.05	25.0	569.4	176.17	226.17
Oxygen		O_2	32.00	1.40	50.3	154.4	29.17	29.92
Propane	C_3	C_3H_8	44.09	1.13	42.5	370.0	68.34	88.68
Propylene	C_{3-}	C_3H_6	42.08	1.15	46.1	365.6	60.16	75.70
Blast furnace gas (1)		–	29.6	1.39	–	–	29.97	30.64
Cat cracker gas (1)		–	28.83	1.20	46.5	286.1	46.16	57.31
Sulfur dioxide		SO_2	64.06	1.24	78.7	430.6	38.05	40.00
Water vapor		H_2O	18.02	1.33	221.2	647.8	33.31	34.07

(Most values taken from Natural Gas Processors Suppliers Association Engineering Data Book-1972, Ninth Edition.)
(1) Approximate values based on average composition.
[a] *Use straight-line interpolation or extrapolation to approximate* Mcp *[in kJ/(kmol K)] at actual inlet* T. *(For greater accuracy,* average T *should be used.)*
Adapted from Elliott multistage centrifugal compressors, Bulletin P-25C. Jeannette, PA: Elliott Co.; 1985.

TABLE A.2 Specific Volume of Various Gases at 14.7 psia, 60°F

Gas	v (ft^3/lb)
Acetylene	14.37
Air	13.09
Ammonia	22.10
Argon	9.5
iso-butane	6.26
n-Butane	6.25
iso-butylene	6.54
Butylene	6.54
Carbon dioxide	8.53
Carbon monoxide	13.55

Continued

TABLE A.2 Specific Volume of Various Gases at 14.7 psia, 60°F—cont'd

Gas	v (ft^3/lb)
Chlorine	5.25
Coke oven gas	34.10
Ethane	12.52
Ethylene	13.4
Flue gas	12.6
Helium	94.91
Hydrogen	187.8
Hydrogen sulfide	11.0
Methane	23.5
Methyl chloride	6.26
Natural gas	20.0
Nitrogen	13.53
Oxygen	11.85
Propane	8.45
Propylene	8.86
Sulfur dioxide	5.8

Source: BWR Gas Properties.

TABLE A.3 Viscosity of Various Gases[a]

	Viscosity (centistokes)		
Gas	**0°F**	**200°F**	**400°F**
Air	0.0160	0.0215	0.0260
Ammonia	0.0088	0.0128	0.0163
Carbon dioxide	0.0130	0.0181	0.0225
Carbon monoxide	0.0155	0.0202	0.0244
Chlorine	0.0122	0.0168	0.0213
Freon-12	0.0115	0.0157	0.0200
Helium	0.0175	0.0228	0.0275
Hydrogen	0.0080	0.0101	0.0121
iso-butane	0.0069	0.0095	0.0120
Methane	0.0085	0.0112	0.0141
Nitrogen	0.0160	0.0208	0.0250
Oxygen	0.0181	0.0240	0.0290
Propane	0.0074	0.0101	0.0128
Sulfur dioxide	0.0108	0.0160	0.0206
Water vapor		0.0107	0.0136

$\nu' =$ Kinematic viscosity, ft^2/s
$= (\text{Viscosity in Centistokes}) \times 1.076 \times 10^{-5}$
$= (\text{Viscosity in Centipoise}) \times 6.72 \times 10^{-4} \times v$

[a]*Approximate values for gas at or near atmospheric pressure and the temperature specified. Note that viscosity is relatively constant with pressure for pressures below 500 psi.*

Source: BWR Gas Properties.

TABLE A.4 Compressor Design Operating Conditions			
Compressor Operating Data			
Barometer inlet capacity	29.98 in. Hg. Abs.		101.35 kPa
Volume flow	11,675 cfm		19835 m^3/h
Mass flow	5039 lb/min		38.3 kg/s
Temperature			
Inlet	122°F		50°C
Discharge	232°F		166.7°C
Maximum discharge	350°F		1.76°C
Minimum	0°F		–17.8°C
1st iso-cool outlet	229°F		109.4°C
1st iso-cool re-entry	110°F		43°C
Pressure			
Inlet	134.7	psia	929 kPa
Rated discharge	573.5	psia	3954 kPa
Maximum discharge	715	psia	4928 kPa
1st iso-cool outlet	270	psia	1862 kPa
1st iso-cool re-entry	264	psia	1822 kPa
Maximum allowable pressure			
Diff. across inter. diaphragm	50	psia	344.75 kPa
Guaranteed power input	14,001	hp	10445 kW
Speed			
Guaranteed	7693	r/min	
Maximum continuous	8077	r/min	
1st critical—book	4835	r/min	
2nd critical—book	11,885	r/min	
Gas inlet conditions			
Molecular mass	19.728		
Specific heat ratio (k_1)	1.247		
Compressibility (Z_1)	0.981		
Gas discharge conditions			
Specific heat ratio (k_2)	1.228		
Compressibility (Z_2)	0.967		

3. The third k value is the isentropic volume exponent. k is calculated by the expression

$$\left(\frac{P_1}{P_2}\right) = \left(\frac{v_2}{v_1}\right)^k$$

where v is the specific volume calculated by the expression $v = ZRT$. This value is used for sonic velocity calculations.

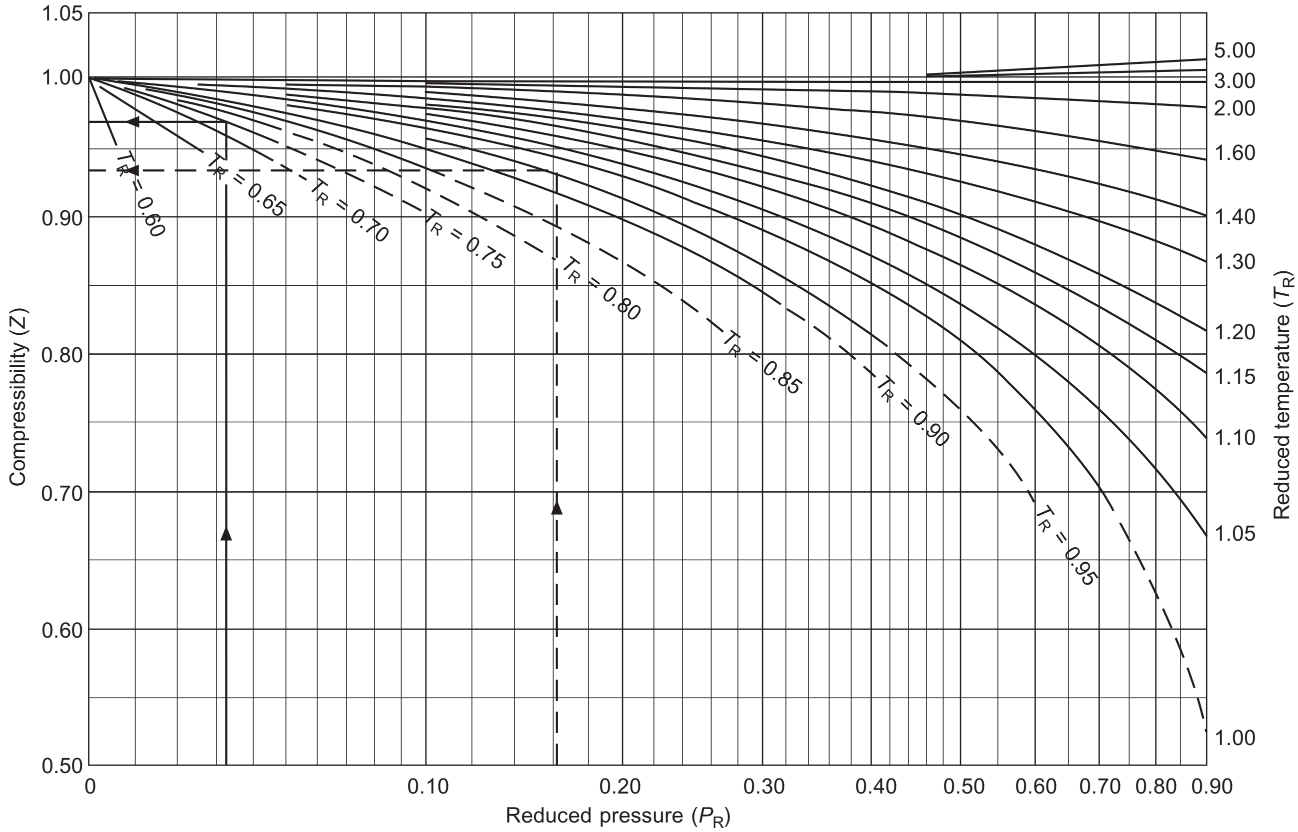


FIG. A.1 Compressibility curve (Obert-Nelson). Used with data from Table A.1A and Eqs. (2.55), (2.56) [30]. *(Used with permission of Elliott Company, Jeannette, PA.)*

Appendix B

Mollier Diagrams

Properties of selected gases are presented in the form of Mollier diagrams (Figs. B.1–B.36). The following gases are included:

Air
Ammonia
Argon
Carbon dioxide
Carbon monoxide
Chlorine
Ethane
Ethylene
Freon-11
Freon-12
Refrigerant 22
Refrigerant 500
Helium
Hydrogen
Iso-butane
Methane
Nitrogen
Oxygen
Propane
Propylene
Steam

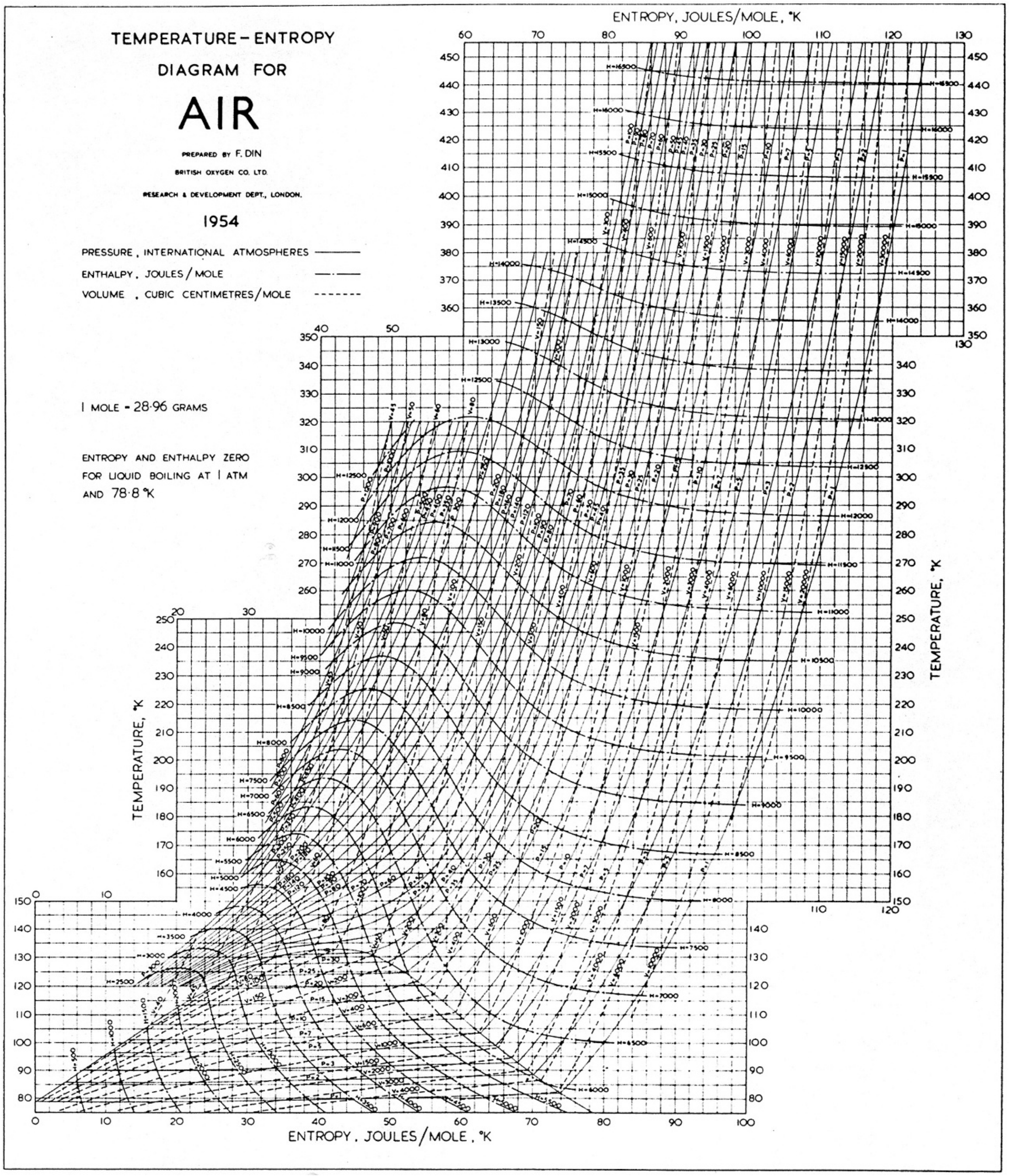


FIG. B.1 Temperature-entropy diagram for air. *(From Din F, editor, Thermodynamic functions of gases, vol. 2, Butterworth & Co. London; 1956, with permission.)*

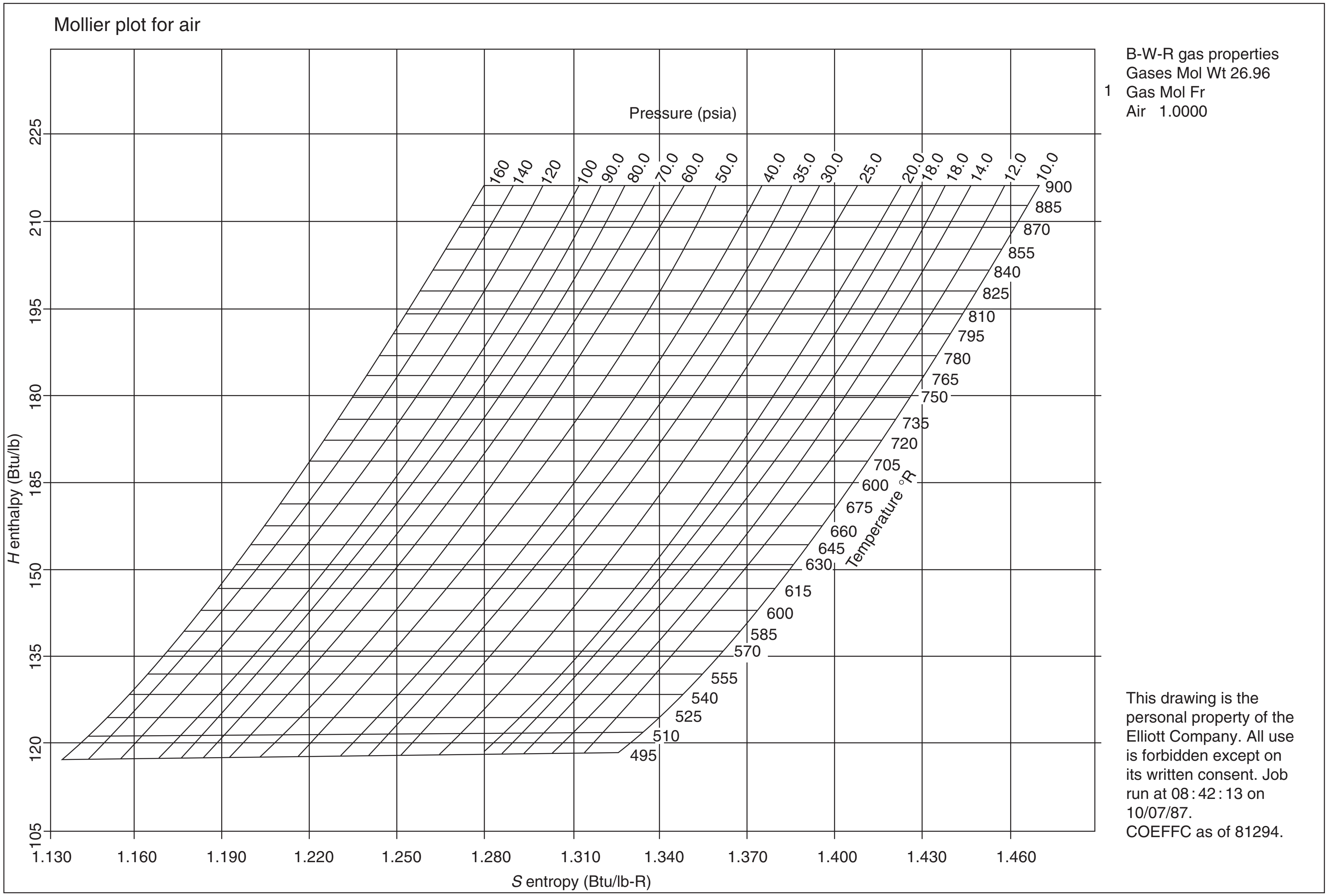


FIG. B.2 Mollier plot for air. *(Used with permission of Elliott Company, Jeannette, PA.)*

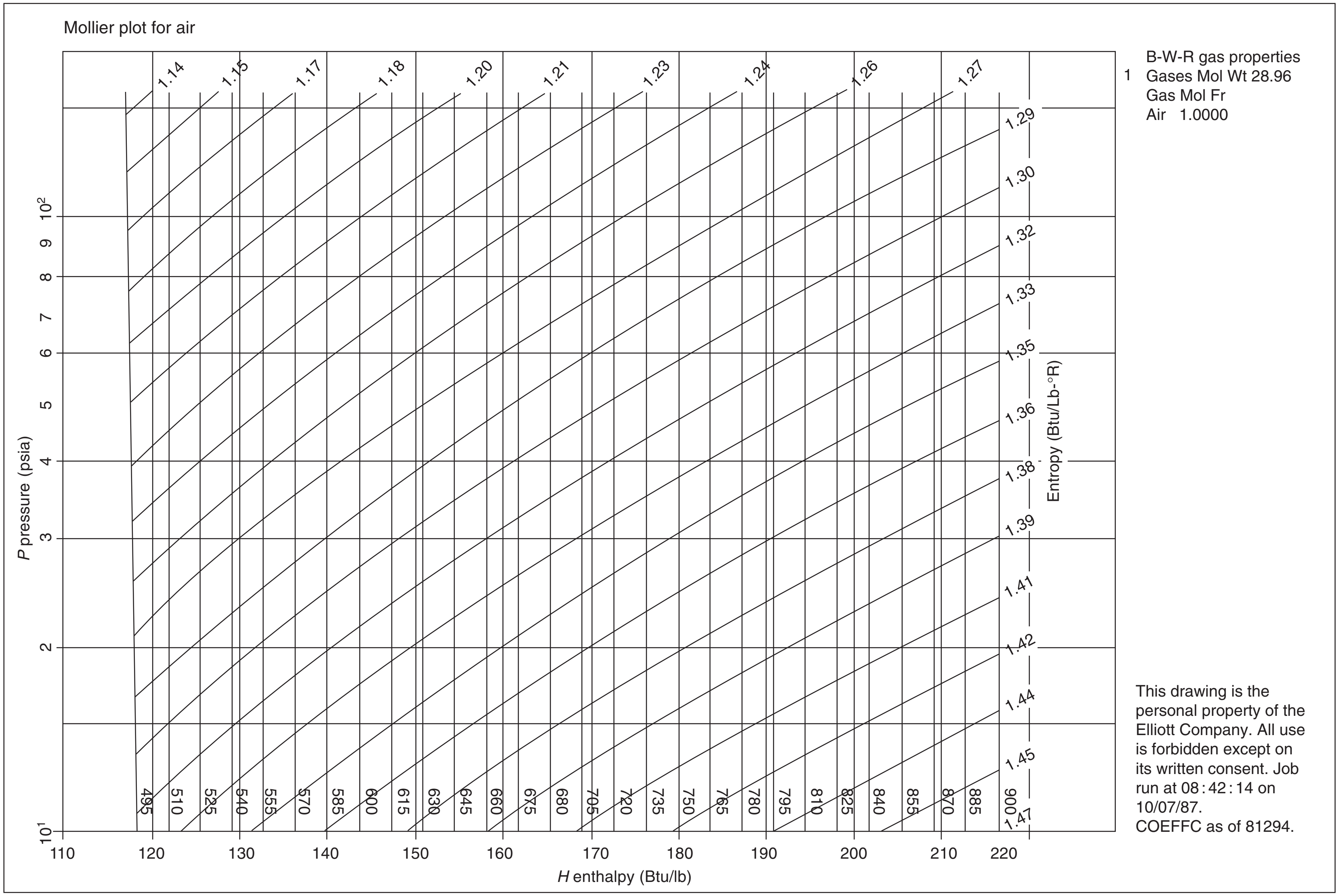


FIG. B.3 Mollier plot for air. *(Used with permission of Elliott Company, Jeannette, PA.)*

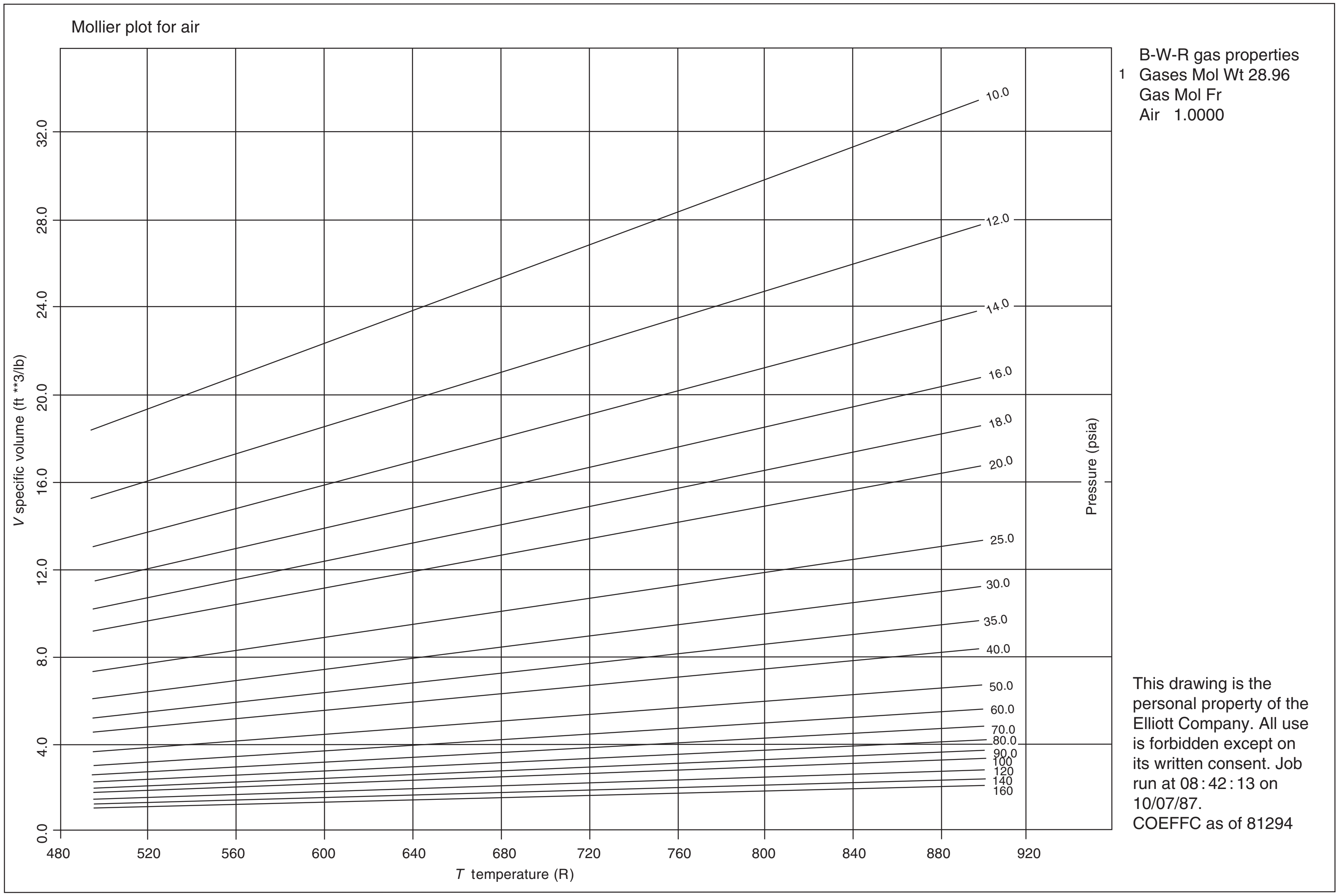


FIG. B.4 Mollier plot for air. *(Used with permission of Elliott Company, Jeannette, PA.)*

Carrier® PSYCHROMETRIC CHART Normal Temperatures

Barometric Pressure
29.92 inches of Mercury

Reproduced by permission of Carrier Corporation.

ADDITIVE CORRECTIONS FOR W, h, AND v WHEN BAROMETRIC PRESSURE DIFFERS FROM STANDARD BAROMETER

Wet-Bulb Temp.	Sat. Vapor Press.	APPROXIMATE ALTITUDE IN FEET													
		−900		900		1800		2700		3700		4800		5900	
		$\Delta p = +1$		$\Delta p = -1$		$\Delta p = -2$		$\Delta p = -3$		$\Delta p = -4$		$\Delta p = -5$		$\Delta p = -6$	
t_{wb}	In. Hg	ΔW_{wb}	Δh	ΔW_{wb}	Δh	ΔW_{wb}	Δh	ΔW_{wb}	Δh	ΔW_{wb}	Δh	ΔW_{wb}	Δh	ΔW_{wb}	Δh
20	.1027	−0.5	−0.08	0.5	0.08	1.1	0.17	1.7	0.26	2.3	0.36	3.0	0.46	3.8	0.58
21	.1078	−0.5	−0.08	0.5	0.08	1.1	0.17	1.8	0.27	2.4	0.37	3.2	0.49	4.0	0.61
22	.1130	−0.5	−0.08	0.6	0.09	1.2	0.18	1.9	0.29	2.6	0.40	3.4	0.52	4.2	0.64
23	.1186	−0.6	−0.09	0.6	0.09	1.3	0.19	2.0	0.30	2.7	0.41	3.6	0.55	4.4	0.67
24	.1243	−0.6	−0.09	0.6	0.10	1.3	0.20	2.1	0.32	2.8	0.43	3.7	0.57	4.6	0.71
25	.1303	−0.6	−0.10	0.7	0.10	1.4	0.21	2.2	0.33	3.0	0.46	3.9	0.60	4.9	0.75
26	.1366	−0.7	−0.10	0.7	0.11	1.4	0.22	2.3	0.35	3.1	0.48	4.1	0.63	5.1	0.78
27	.1431	−0.7	−0.11	0.7	0.11	1.5	0.23	2.4	0.37	3.2	0.50	4.3	0.66	5.4	0.83
28	.1500	−0.7	−0.11	0.8	0.12	1.6	0.24	2.5	0.38	3.4	0.52	4.5	0.69	5.6	0.86
29	.1571	−0.8	−0.12	0.8	0.12	1.7	0.26	2.6	0.40	3.6	0.55	4.7	0.72	5.8	0.89
30	.1645	−0.8	−0.12	0.8	0.13	1.7	0.27	2.7	0.42	3.8	0.58	4.9	0.75	6.1	0.92
31	.1723	−0.8	−0.13	0.9	0.13	1.8	0.28	2.8	0.44	3.9	0.60	5.1	0.78	6.3	0.95
32	.1803	−0.9	−0.13	0.9	0.14	1.9	0.29	3.0	0.45	4.1	0.63	5.3	0.82	6.6	1.01
33	.1878	−0.9	−0.14	1.0	0.15	2.0	0.30	3.1	0.47	4.3	0.66	5.5	0.85	6.9	1.06
34	.1955	−0.9	−0.14	1.0	0.15	2.1	0.32	3.2	0.49	4.4	0.68	5.7	0.88	7.2	1.11
35	.2034	−1.0	−0.15	1.0	0.16	2.1	0.33	3.4	0.51	4.6	0.71	6.0	0.92	7.5	1.16
36	.2117	−1.0	−0.15	1.1	0.17	2.2	0.35	3.5	0.53	4.8	0.74	6.2	0.96	7.8	1.20
37	.2202	−1.0	−0.16	1.1	0.17	2.3	0.36	3.6	0.56	5.0	0.77	6.5	1.00	8.1	1.25
38	.2290	−1.1	−0.17	1.2	0.18	2.4	0.37	3.8	0.58	5.2	0.80	6.8	1.05	8.4	1.30
39	.2382	−1.1	−0.18	1.2	0.19	2.5	0.39	3.9	0.61	5.5	0.85	7.1	1.09	8.8	1.36
40	.2477	−1.2	−0.18	1.3	0.20	2.6	0.41	4.1	0.63	5.7	0.88	7.4	1.14	9.2	1.42
41	.2575	−1.2	−0.19	1.3	0.20	2.7	0.42	4.3	0.66	5.9	0.91	7.7	1.19	9.6	1.48
42	.2676	−1.3	−0.20	1.4	0.21	2.8	0.44	4.4	0.69	6.1	0.94	8.0	1.23	10.0	1.54
43	.2781	−1.3	−0.21	1.4	0.22	3.0	0.45	4.6	0.71	6.4	0.99	8.4	1.29	10.4	1.61
44	.2890	−1.4	−0.22	1.5	0.23	3.1	0.47	4.8	0.74	6.7	1.04	8.7	1.34	10.8	1.67
45	.3002	−1.4	−0.22	1.6	0.24	3.2	0.49	5.0	0.77	6.9	1.07	9.1	1.40	11.3	1.75
46	.3119	−1.5	−0.23	1.6	0.25	3.3	0.51	5.2	0.80	7.2	1.11	9.4	1.45	11.7	1.81
47	.3239	−1.6	−0.24	1.7	0.26	3.4	0.53	5.4	0.84	7.5	1.16	9.8	1.52	12.1	1.87
48	.3363	−1.6	−0.25	1.8	0.27	3.6	0.56	5.6	0.87	7.8	1.21	10.2	1.58	12.6	1.95
49	.3491	−1.7	−0.26	1.8	0.28	3.7	0.58	5.8	0.90	8.1	1.25	10.5	1.63	13.1	2.03
50	.3624	−1.7	−0.27	1.9	0.29	3.9	0.60	6.1	0.94	8.4	1.30	10.9	1.69	13.6	2.11
51	.3761	−1.8	−0.28	2.0	0.30	4.0	0.63	6.3	0.97	8.7	1.35	11.3	1.75	14.1	2.18
52	.3903	−1.9	−0.29	2.0	0.32	4.2	0.65	6.5	1.01	9.0	1.40	11.8	1.83	14.7	2.28
53	.4049	−1.9	−0.30	2.1	0.33	4.4	0.68	6.7	1.05	9.3	1.44	12.2	1.89	15.2	2.36
54	.4200	−2.0	−0.31	2.2	0.34	4.5	0.70	7.0	1.09	9.7	1.50	12.7	1.97	15.8	2.45
55	.4356	−2.1	−0.32	2.3	0.35	4.7	0.73	7.3	1.13	10.1	1.57	13.2	2.05	16.4	2.54
56	.4518	−2.2	−0.34	2.4	0.37	4.9	0.76	7.6	1.18	10.5	1.63	13.7	2.13	17.1	2.66
57	.4684	−2.3	−0.35	2.4	0.37	5.1	0.79	7.9	1.22	10.9	1.69	14.2	2.21	17.7	2.75
58	.4856	−2.3	−0.37	2.5	0.39	5.3	0.82	8.2	1.27	11.3	1.76	14.7	2.28	18.4	2.86
59	.5033	−2.4	−0.38	2.6	0.41	5.4	0.85	8.5	1.32	11.7	1.82	15.3	2.38	19.1	2.97
60	.5216	−2.5	−0.40	2.7	0.42	5.7	0.88	8.8	1.37	12.2	1.90	15.9	2.47	19.9	3.09
61	.5405	−2.6	−0.41	2.8	0.44	5.9	0.91	9.2	1.43	12.7	1.98	16.5	2.57	20.7	3.22
62	.5599	−2.7	−0.43	2.9	0.46	6.1	0.95	9.5	1.48	13.2	2.05	17.1	2.66	21.4	3.33
63	.5800	−2.8	−0.44	3.0	0.48	6.3	0.98	9.9	1.54	13.7	2.13	17.7	2.76	22.3	3.47
64	.6007	−2.9	−0.46	3.2	0.49	6.5	1.02	10.2	1.59	14.2	2.21	18.4	2.87	23.1	3.60
65	.6221	−3.1	−0.48	3.3	0.51	6.8	1.06	10.6	1.65	14.7	2.29	19.1	2.98	23.9	3.73
66	.6441	−3.2	−0.50	3.4	0.53	7.1	1.10	11.0	1.72	15.3	2.38	19.8	3.09	24.8	3.87
67	.6668	−3.3	−0.51	3.5	0.55	7.3	1.14	11.4	1.78	15.8	2.47	20.5	3.20	25.7	4.01
68	.6902	−3.4	−0.53	3.7	0.57	7.6	1.18	11.8	1.84	16.4	2.56	21.3	3.32	26.7	4.16
69	.7143	−3.5	−0.55	3.8	0.59	7.9	1.23	12.2	1.90	17.0	2.65	22.1	3.45	27.7	4.32
70	.7392	−3.7	−0.57	3.9	0.61	8.1	1.27	12.7	1.98	17.6	2.75	22.9	3.58	28.7	4.48
71	.7648	−3.8	−0.59	4.1	0.64	8.4	1.32	13.1	2.05	18.2	2.84	23.7	3.70	29.7	4.64
72	.7911	−3.9	−0.61	4.2	0.66	8.7	1.36	13.6	2.13	18.8	2.94	24.6	3.84	30.9	4.82
73	.8183	−4.1	−0.63	4.4	0.69	9.0	1.41	14.1	2.20	19.5	3.05	25.5	3.99	31.9	4.99
74	.8463	−4.2	−0.66	4.6	0.71	9.4	1.46	14.6	2.28	20.2	3.16	26.4	4.14	33.1	5.18
75	.8750	−4.4	−0.68	4.7	0.74	9.7	1.52	15.1	2.36	20.9	3.27	27.4	4.28	34.3	5.37
76	.9047	−4.5	−0.71	4.9	0.77	10.0	1.57	15.7	2.46	21.7	3.39	28.3	4.42	35.5	5.56
77	.9352	−4.7	−0.73	5.1	0.79	10.4	1.63	16.3	2.55	22.5	3.52	29.4	4.61	36.9	5.77
78	.9667	−4.9	−0.76	5.2	0.82	10.8	1.69	16.9	2.65	23.3	3.65	30.5	4.77	38.2	5.98
79	.9990	−5.0	−0.79	5.4	0.85	11.2	1.75	17.5	2.74	24.2	3.79	31.6	4.95	39.6	6.20
80	1.032	−5.2	−0.82	5.6	0.88	11.6	1.82	18.1	2.84	25.1	3.93	32.7	5.13	41.0	6.43
81	1.067	−5.4	−0.85	5.8	0.91	12.0	1.88	18.8	2.95	26.0	4.08	33.9	5.32	42.5	6.66
82	1.102	−5.6	−0.88	6.0	0.94	12.5	1.96	19.5	3.06	27.0	4.24	35.1	5.51	44.0	6.90
83	1.138	−5.8	−0.91	6.2	0.97	12.9	2.02	20.2	3.17	28.0	4.39	36.4	5.71	45.6	7.15
84	1.175	−6.0	−0.94	6.4	1.00	13.3	2.10	20.9	3.28	28.9	4.54	37.7	5.92	47.2	7.41

t_{db} = Dry-bulb temperature (F)
t_{wb} = Wet-bulb temperature (F)
p = Barometric pressure (in. of Hg)
$\Delta p'$ = Pressure difference from standard barometer (in. of Hg)
W = Moisture content of air (gr per lb of dry air)
W_{wb} = Moisture content of air saturated at wet-bulb temperature t_{wb} (gr per lb of dry air)
ΔW = Moisture content correction of air when barometric pressure differs from standard barometer (gr per lb of dry air)
ΔW_{wb} = Moisture content correction of air saturated at wet-bulb temperature when barometric pressure differs from standard barometer (gr per lb of dry air)

NOTE: To obtain ΔW reduce ΔW_{wb} by 1% where $t_{db} - t_{wb} = 24$ F

Correct proportionally, when $t_{db} - t_{wb}$ = is not 24 F

h = Enthalpy of moist air (Btu per lb of dry air)
Δh = Enthalpy correction, for saturated or unsaturated air, when barometric pressure differs from standard (Btu per lb of dry air)
= Volume of moist air (cu ft per lb of dry air)

$$v = \frac{.754\,(t_{db} + 459.7)}{p}\left[1 + \frac{W}{4360}\right]$$

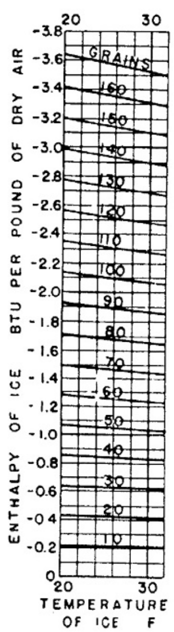
Example: At a barometric pressure of 25.92 with 90 F DB and 70 F WB, determine W, h, and v. $\Delta p = -4$ and from table $\Delta W_{wb} = 17.6$. From note above,

$$\Delta W = \Delta W_{wb} - \left(\frac{20}{24} \times .01 \times 17.6\right) = 17.6 - .15 = 17.45$$

Therefore W = 78.0 (from chart) + 17.45 = 95.45 gr per lb of dry air. From table Δh = 2.75. Therefore h = saturation enthalpy from chart + deviation + 2.75 = 34.09 − .18 + 2.75 = 36.66 Btu per lb of dry air. From equation above

$$v = \frac{.754\,(90 + 459.7)}{25.92}\left[1 + \frac{95.45}{4360}\right] = 16.34 \text{ cu ft per lb of dry air}$$

ENTHALPY OF ADDED OR REJECTED MOISTURE



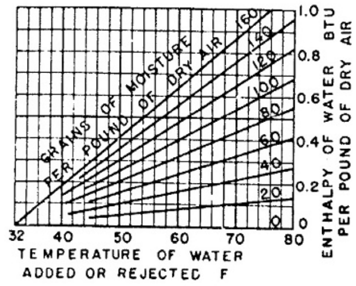


FIG. B.5 Psychometric chart for normal temperatures. *(Reproduced by permission of Carrier Corp.)*

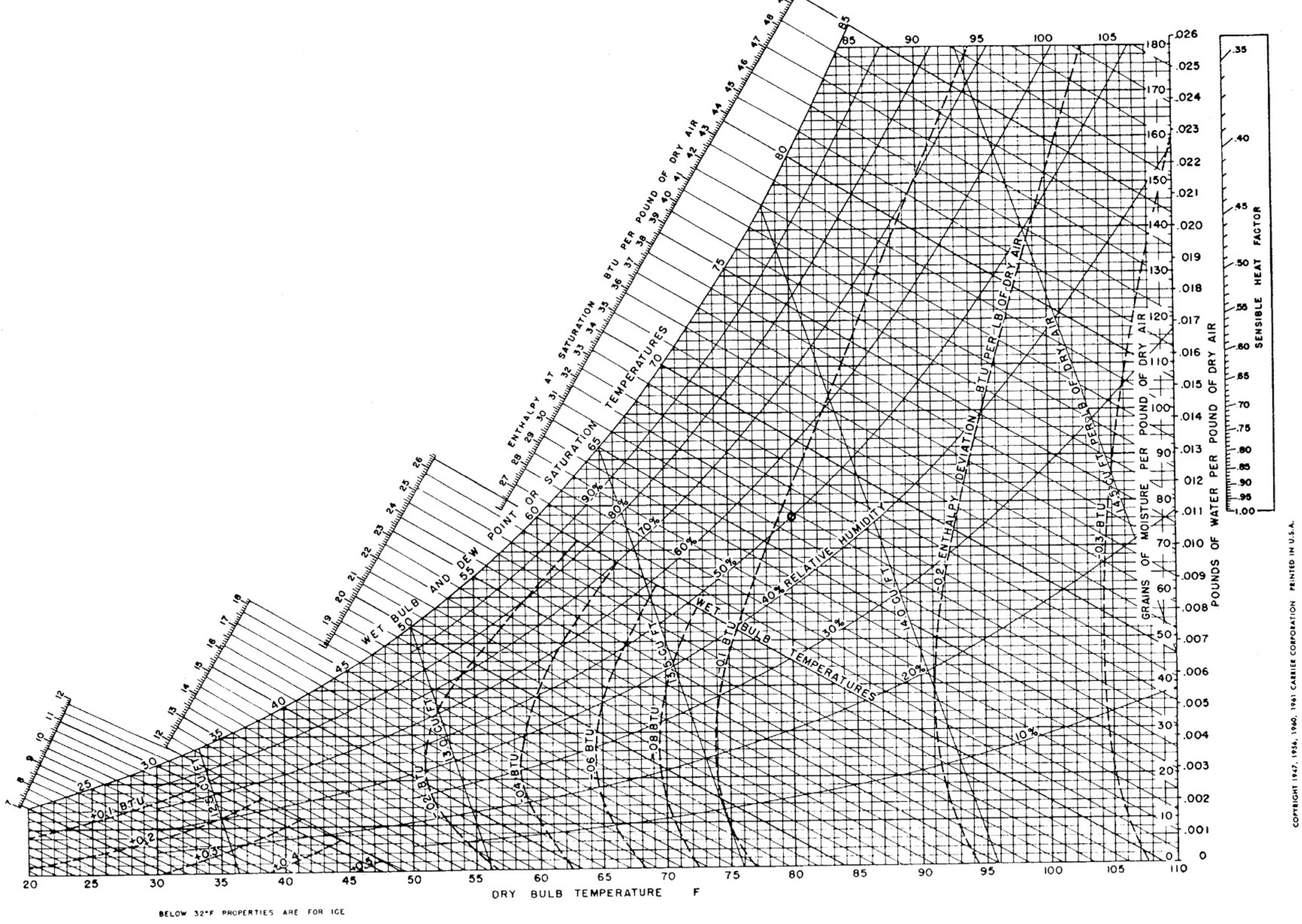

ENTHALPY AT SATURATION BTU PER POUND OF DRY AIR
WET BULB AND DEW POINT OR SATURATION TEMPERATURES
RELATIVE HUMIDITY
WET BULB TEMPERATURES
ENTHALPY DEVIATION BTU PER LB OF DRY AIR
CU FT PER LB OF DRY AIR
GRAINS OF MOISTURE PER POUND OF DRY AIR
POUNDS OF WATER PER POUND OF DRY AIR
SENSIBLE HEAT FACTOR
DRY BULB TEMPERATURE F
BELOW 32°F PROPERTIES ARE FOR ICE
COPYRIGHT 1947, 1956, 1960, 1961 CARRIER CORPORATION PRINTED IN U.S.A.

FIG. B.5, CONT'D

Carrier®

Reproduced by permission of Carrier Corporation.

PSYCHROMETRIC CHART
LOW TEMPERATURES

Barometric Pressure 29.92 In. Hg

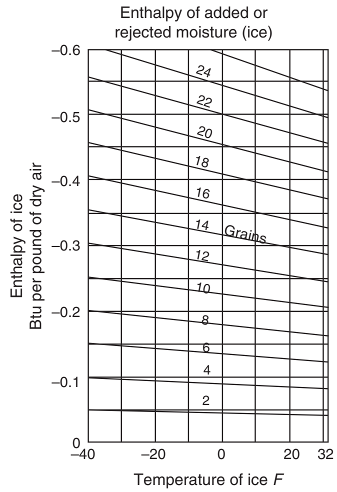
ADDITIVE CORRECTIONS FOR W, h, AND v WHEN BAROMETRIC PRESSURE differs from standard barometer

Wet-Bulb temp. t_{wb}	Sat. vapor press. in Hg	Approximate altitude in feet													
		–900		900		1800		2700		3700		4800		5900	
		$\Delta p = +1$		$\Delta p = -1$		$\Delta p = -2$		$\Delta p = -3$		$\Delta p = -4$		$\Delta p = -5$		$\Delta p = -6$	
		ΔW_{wb}	Δh	ΔW_{wb}	Δh	ΔW_{wb}	Δh	ΔW_{wb}	Δh	ΔW_{wb}	Δh	ΔW_{wb}	Δh	ΔW_{wb}	Δh
–20	0.013	–0.06	–0.01	0.06	0.01	0.13	0.02	0.20	0.03	0.28	0.04	0.36	0.05	0.47	0.07
–18	0.014	–0.07	–0.01	0.07	0.01	0.14	0.02	0.23	0.03	0.32	0.05	0.41	0.06	0.52	0.08
–16	0.016	–0.07	–0.01	0.08	0.01	0.16	0.02	0.26	0.04	0.36	0.05	0.46	0.07	0.58	0.09
–14	0.018	–0.08	–0.01	0.09	0.01	0.18	0.03	0.29	0.04	0.40	0.06	0.52	0.08	0.65	0.10
–12	0.020	–0.09	–0.01	0.10	0.01	0.21	0.03	0.32	0.05	0.44	0.07	0.58	0.09	0.72	0.11
–10	0.022	–0.10	–0.02	0.11	0.02	0.23	0.03	0.35	0.05	0.50	0.07	0.64	0.10	0.81	0.12
–8	0.025	–0.12	–0.02	0.12	0.02	0.26	0.04	0.40	0.06	0.55	0.08	0.72	0.11	0.90	0.13
–6	0.027	–0.13	–0.02	0.14	0.02	0.29	0.04	0.44	0.07	0.62	0.09	0.80	0.12	1.00	0.15
–4	0.030	–0.14	–0.02	0.15	0.02	0.32	0.05	0.50	0.07	0.69	0.10	0.89	0.13	1.12	0.17
–2	0.034	–0.16	–0.02	0.17	0.02	0.35	0.05	0.55	0.08	0.76	0.11	0.99	0.15	1.24	0.19
0	0.038	–0.18	–0.03	0.19	0.03	0.39	0.06	0.61	0.09	0.85	0.13	1.10	0.17	1.38	0.21
2	0.042	–0.20	–0.03	0.21	0.03	0.44	0.07	0.68	0.10	0.94	0.14	1.22	0.19	1.53	0.23
4	0.046	–0.22	–0.03	0.23	0.03	0.48	0.07	0.75	0.11	1.05	0.16	1.36	0.21	1.70	0.26
6	0.051	–0.24	–0.04	0.26	0.04	0.54	0.08	0.83	0.13	1.16	0.18	1.51	0.23	1.89	0.29
8	0.057	–0.27	–0.04	0.29	0.04	0.59	0.09	0.93	0.14	1.28	0.19	1.67	0.25	2.09	0.32
10	0.063	–0.30	–0.04	0.32	0.05	0.66	0.10	1.03	0.16	1.42	0.22	1.85	0.28	2.31	0.35
12	0.069	–0.33	–0.05	0.35	0.05	0.73	0.11	1.13	0.17	1.57	0.24	2.04	0.31	2.56	0.39
14	0.077	–0.36	–0.05	0.39	0.06	0.81	0.12	1.25	0.19	1.74	0.26	2.26	0.34	2.82	0.43
16	0.085	–0.40	–0.06	0.43	0.06	0.89	0.14	1.38	0.21	1.92	0.29	2.49	0.38	3.12	0.48
18	0.093	–0.44	–0.07	0.47	0.07	0.98	0.15	1.53	0.23	2.12	0.32	2.75	0.42	3.44	0.53
20	0.103	–0.49	–0.08	0.52	0.08	1.08	0.17	1.68	0.26	2.33	0.36	3.03	0.46	3.79	0.58
22	0.113	–0.5	–0.08	0.6	0.09	1.2	0.18	1.9	0.29	2.6	0.40	3.4	0.52	4.2	0.64
24	0.124	–0.6	–0.09	0.6	0.10	1.3	0.20	2.1	0.32	2.8	0.43	3.7	0.57	4.6	0.71
26	0.137	–0.7	–0.10	0.7	0.11	1.4	0.22	2.3	0.35	3.1	0.48	4.1	0.63	5.1	0.78
28	0.150	–0.7	–0.11	0.8	0.12	1.6	0.24	2.5	0.38	3.4	0.52	4.5	0.69	5.6	0.86
30	0.165	–0.8	–0.12	0.8	0.13	1.7	0.27	2.7	0.42	3.8	0.58	4.9	0.75	6.1	0.92
32	0.180	–0.9	–0.13	0.9	0.14	1.9	0.29	3.0	0.45	4.1	0.63	5.3	0.82	6.6	1.01
34	0.197	–0.9	–0.14	1.0	0.15	2.1	0.32	3.2	0.49	4.4	0.68	5.7	0.88	7.2	1.11
36	0.212	–1.0	–0.15	1.1	0.17	2.2	0.35	3.5	0.53	4.8	0.74	6.2	0.96	7.8	1.20
38	0.229	–1.1	–0.17	1.2	0.18	2.4	0.37	3.8	0.58	5.2	0.80	6.8	1.05	8.4	1.30
40	0.248	–1.2	–0.18	1.3	0.20	2.6	0.41	4.1	0.63	5.7	0.88	7.4	1.14	9.2	1.42
42	0.268	–1.3	–0.20	1.4	0.21	2.8	0.44	4.4	0.69	6.1	0.94	8.0	1.23	10.0	1.54
44	0.289	–1.4	–0.22	1.5	0.23	3.1	0.47	4.8	0.74	6.7	1.04	8.7	1.34	10.8	1.67
46	0.312	–1.5	–0.23	1.6	0.25	3.3	0.51	5.2	0.80	7.2	1.11	9.4	1.45	11.7	1.81
48	0.336	–1.6	–0.25	1.8	0.27	3.6	0.56	5.6	0.87	7.8	1.21	10.2	1.58	12.6	1.95

Properties of saturated air
–20 to –100 F

Temp. F	Vapor press. in. Hg	Moisture content grains*	Volume cut ft*	Enthalpy Btu*
–20	.0126	1.83	11.08	–4.52
–22	.0112	1.63	11.03	–5.03
–24	.0100	1.45	10.99	–5.54
–26	.0089	1.29	10.93	–6.04
–28	.0079	1.15	10.88	–6.55
–30	.0070	1.02	10.83	–7.05
–32	.0062	0.90	10.78	–7.54
–34	.0055	0.80	10.73	–8.04
–36	.0049	0.71	10.68	–8.53
–38	.0043	0.62	10.63	–9.03
–40	.0038	0.55	10.58	–9.52
–42	.0033	0.49	10.53	–10.01
–44	.0029	0.43	10.48	–10.50
–46	.0026	0.38	10.43	–10.98
–48	.0023	0.33	10.38	–11.47
–50	.0020	0.29	10.33	–11.96
–52	.0017	0.25	10.28	–12.44
–54	.0015	0.22	10.23	–12.93
–56	.0013	0.19	10.18	–13.41
–58	.0012	0.17	10.13	–13.89
–60	.0010	0.15	10.08	–14.38
–62	.0009	0.13	10.03	–14.86
–64	.0008	0.11	9.98	–15.34
–66	.0007	0.10	9.93	–15.83
–68	.0006	0.08	9.87	–16.31
–70	.0005	0.07	9.82	–16.79
–72	.0004	0.06	9.77	–17.27
–74	.0004	0.05	9.72	–17.75
–76	.0003	0.05	9.67	–18.23
–78	.0003	0.04	9.62	–18.71
–80	.0002	0.03	9.57	–19.19
–82	.0002	0.03	9.52	–19.68
–84	.0002	0.03	9.47	–20.16
–86	.0001	0.02	9.42	–20.64
–88	.0001	0.02	9.37	–21.12
–90	.0001	0.02	9.32	–21.60
–92	.0001	0.01	9.27	–22.80
–94	.0001	0.01	9.22	–22.56
–96	.0001	0.01	9.17	–23.04
–98	.0001	0.01	9.12	–23.52
–100	.0000	0.01	9.07	–24.00

*Per lb of dry air



t_{db} = Dry-bulb temperature (F).
t_{wb} = Wet-bulb temperature (F).
p = Barometric pressure (in. of Hg).
Δp = Pressure difference from standard barometer (in. of Hg).
W = Moisture content of air (gr per lb of dry air).
W_{wb} = Moisture content of air saturated at wet-bulb temperature t_{wb} (gr per lb of dry air).
ΔW = Moisture content correction of air when barometric pressure differs from standard barometer (gr per lb of dry air).
ΔW_{wb} = Moisture content correction of air saturated at wet-bulb temperature when barometric pressure differs from standard barometer (gr per lb of dry air).
Note: To obtain ΔW reduce value of ΔW_{wb} by 1% where $t - t_{wb} = 24$ F and correct proportionally when $t - t_{wb}$ is not 24 F.
h = Enthalpy of moist air (Btu per lb of dry air).
Δh = Enthalpy correction when barometer pressure differs from standard barometer, for saturated or unsaturated air (Btu per lb of dry air).
v = Volume of moist air (cu ft per lb of dry air)

$$v = \frac{.754(t_{db} + 459.7)}{p}\left[1 + \frac{W}{4360}\right]$$

Example: At a barometric pressure of 25.92 with 30 F DB and 28 F WB, determine W, h, and v. $\Delta p = -4$ and from table $\Delta W_{wb} = 3.4$. From note above,

$$\Delta W = \Delta W_{wb} - \left(\frac{2}{24} \times .01 \times 3.4\right) = 3.4 - .003 = 3.4.$$

Therefore $W = 19.2$ (from chart) + 3.4 = 22.6 gr per lb of dry air. From table $\Delta h = 0.52$. Therefore h = saturation enthalpy from chart + deviation + 0.52 = 10.1 + .06 + 0.52 = 10.68 Btu per lb of dry air. From equation above

$$v = \frac{.754(30 + 459.7)}{25.92}\left[1 + \frac{22.6}{4360}\right] = 14.32 \text{ cu ft per lb of dry air.}$$

FIG. B.6 Psychometric chart for low temperatures. *(Reproduced by permission of Carrier Corp.)*

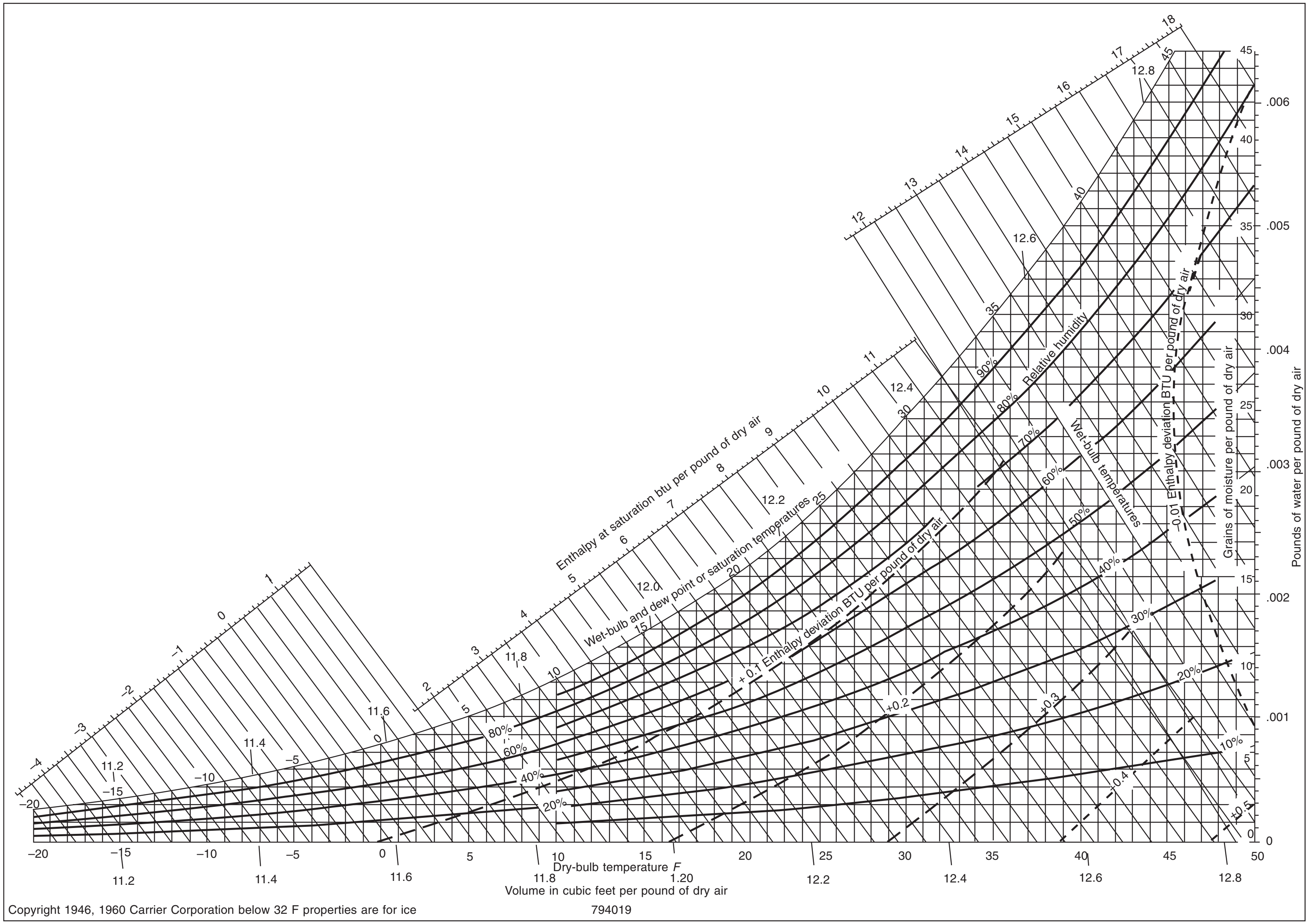

Enthalpy at saturation btu per pound of dry air
Wet-bulb and dew point or saturation temperatures
Relative humidity
Wet-bulb temperatures
Enthalpy deviation BTU per pound of dry air
−0.01 Enthalpy deviation BTU per pound of dry air
Grains of moisture per pound of dry air
Pounds of water per pound of dry air
Dry-bulb temperature F
Volume in cubic feet per pound of dry air
Copyright 1946, 1960 Carrier Corporation below 32 F properties are for ice
794019

FIG. B.6, CONT'D

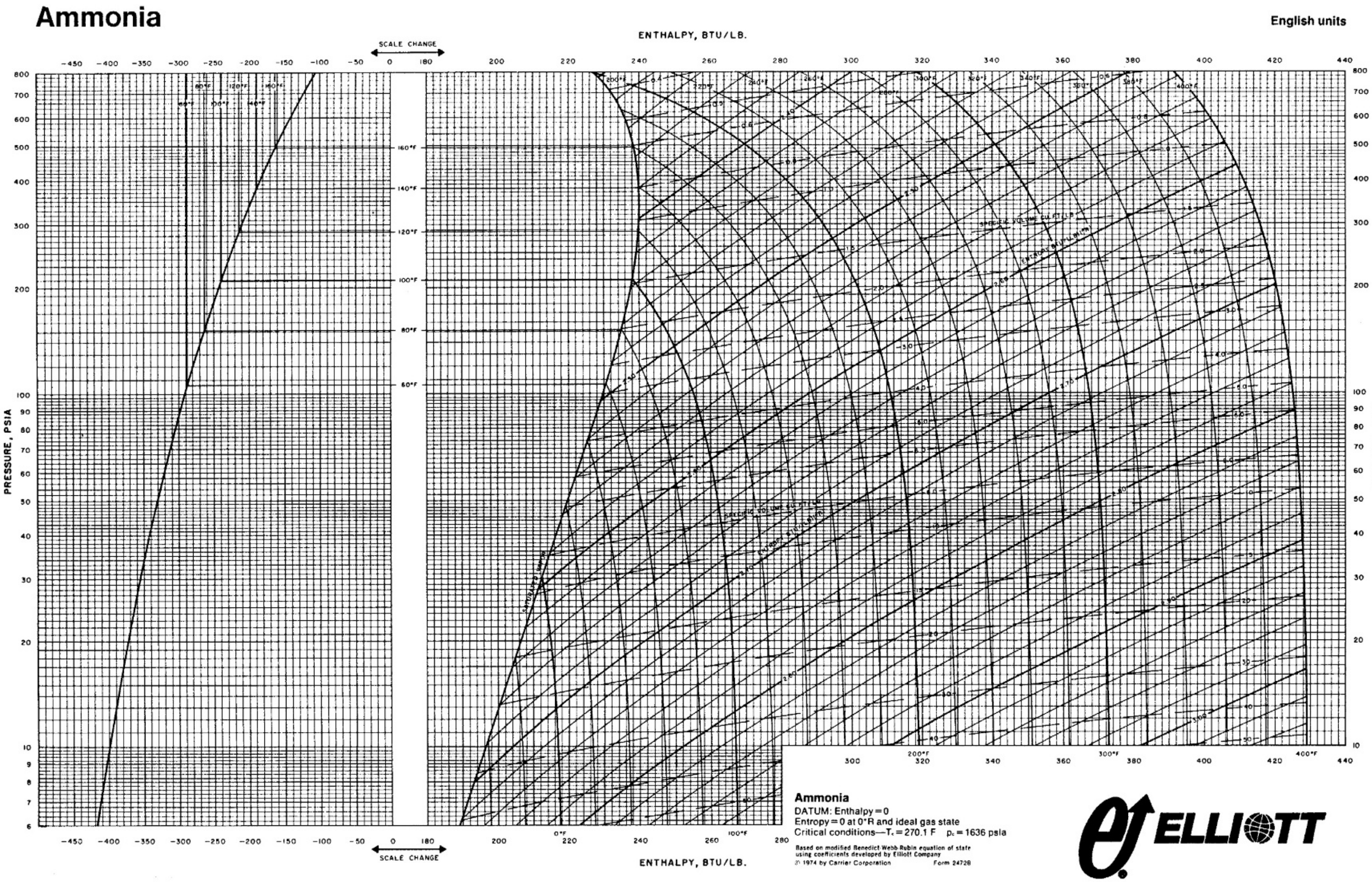


FIG. B.7 Mollier plot for ammonia. *(Used with permission of Elliott Company, Jeannette, PA.)*

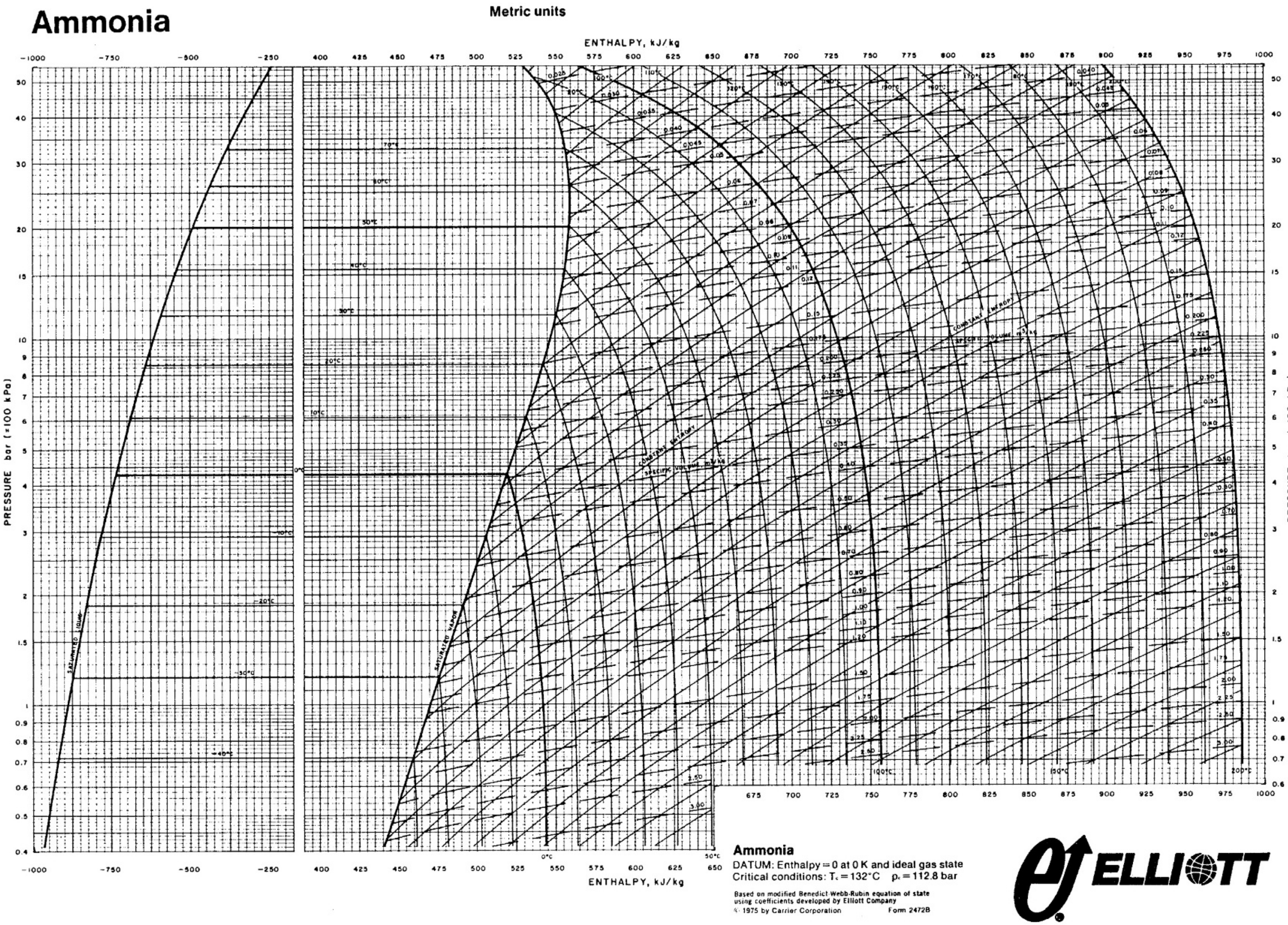


FIG. B.8 Mollier plot for ammonia. *(Used with permission of Elliott Company, Jeannette, PA.)*

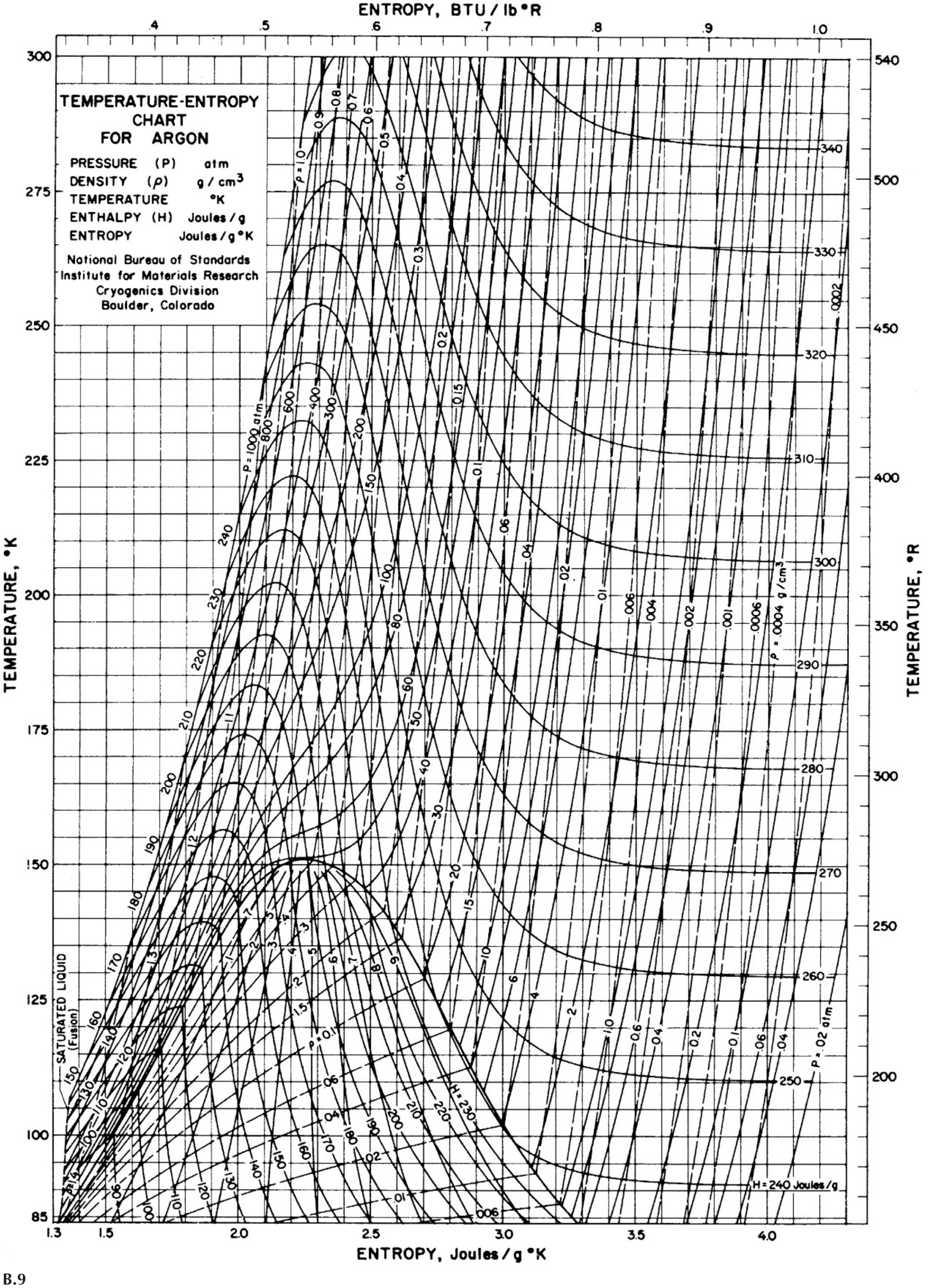

ENTROPY, BTU / lb °R
TEMPERATURE-ENTROPY CHART FOR ARGON
PRESSURE (P) atm
DENSITY (ρ) g / cm³
TEMPERATURE °K
ENTHALPY (H) Joules / g
ENTROPY Joules / g °K
National Bureau of Standards
Institute for Materials Research
Cryogenics Division
Boulder, Colorado
TEMPERATURE, °K
TEMPERATURE, °R
ENTROPY, Joules / g °K
SATURATED LIQUID (Fusion)
H = 240 Joules / g
P = 1000 atm
ρ = .0004 g / cm³
P = .02 atm

FIG. B.9

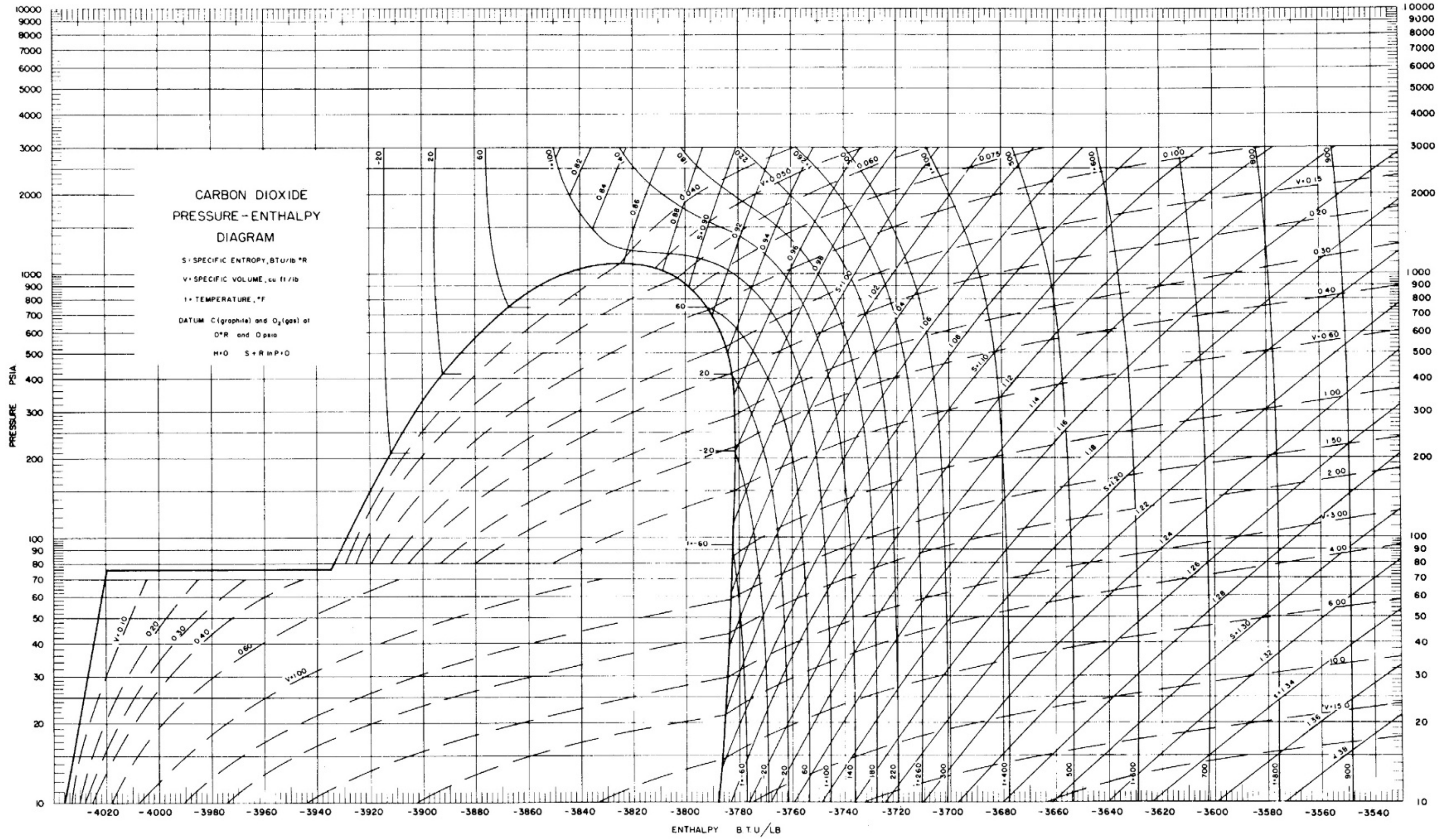


FIG. B.10 Pressure Enthalpy chart for Carbon Dioxide. *(From reference Engineering data book. Tulsa, OK: Natural Gas Processors Suppliers Association; 1967, 1972. Thermo properties of non-hydrocarbons, by Canjar LN, Pollock EK, Cadman TW, Lee WE, Manning FS. Copyright © 1966 by Gulf Publishing Company, Houston, TX. Used with permission. All rights reserved.)*

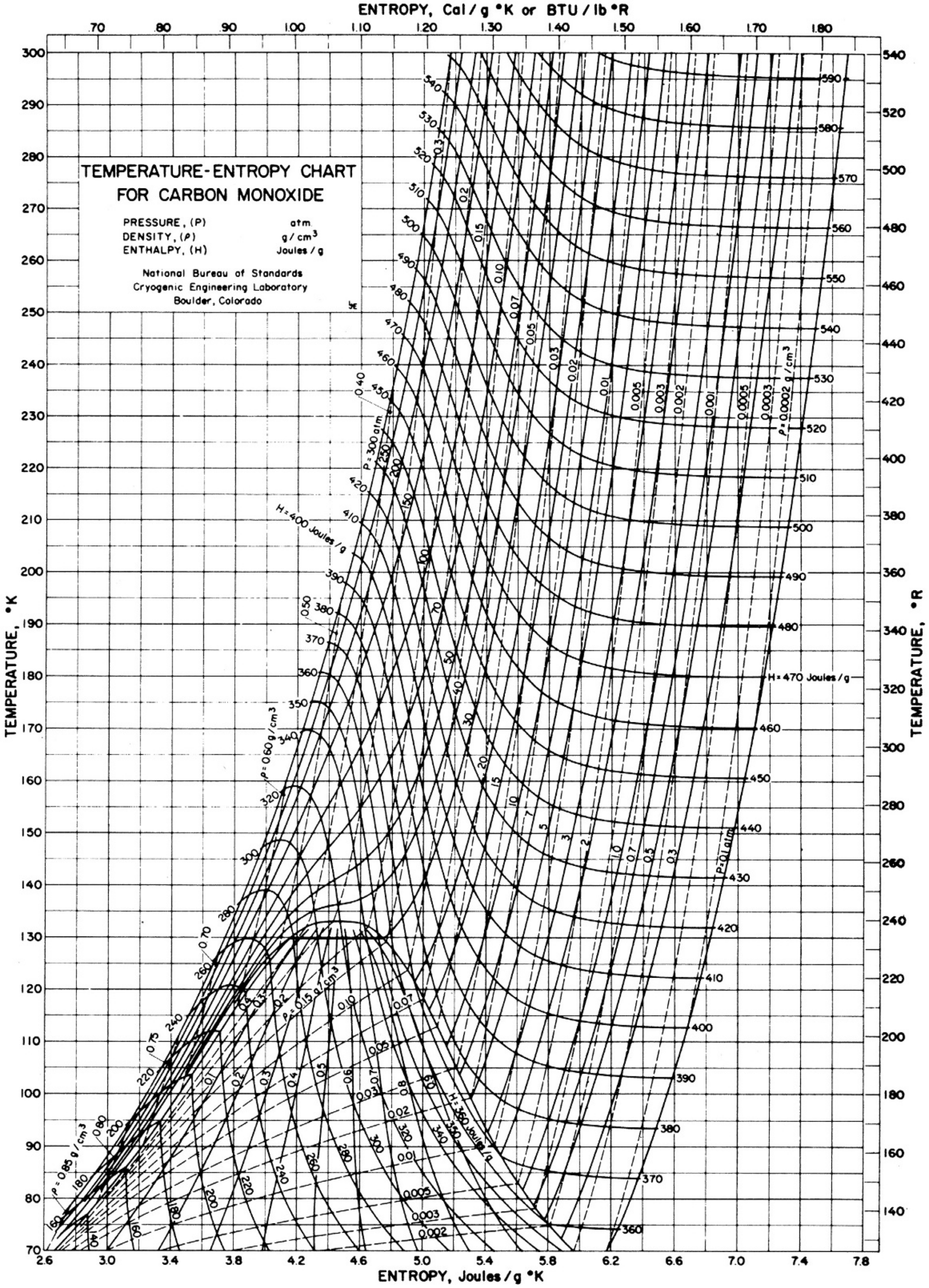


FIG. B.11 Temperature-entropy chart for carbon monoxide. *(From the National Bureau of Standards, Boulder, CO.)*

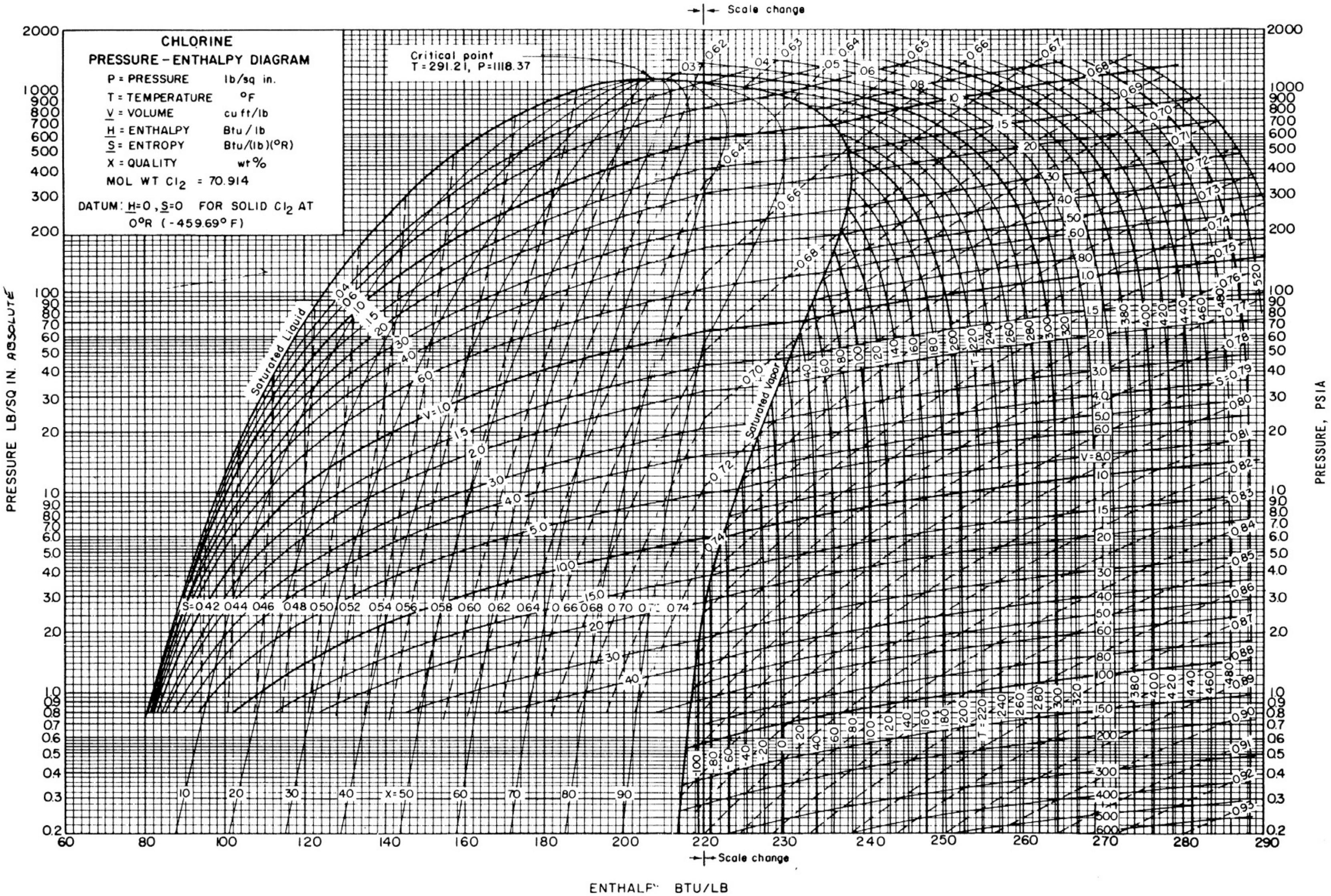


FIG. B.12 Pressure enthalpy diagram for chlorine.

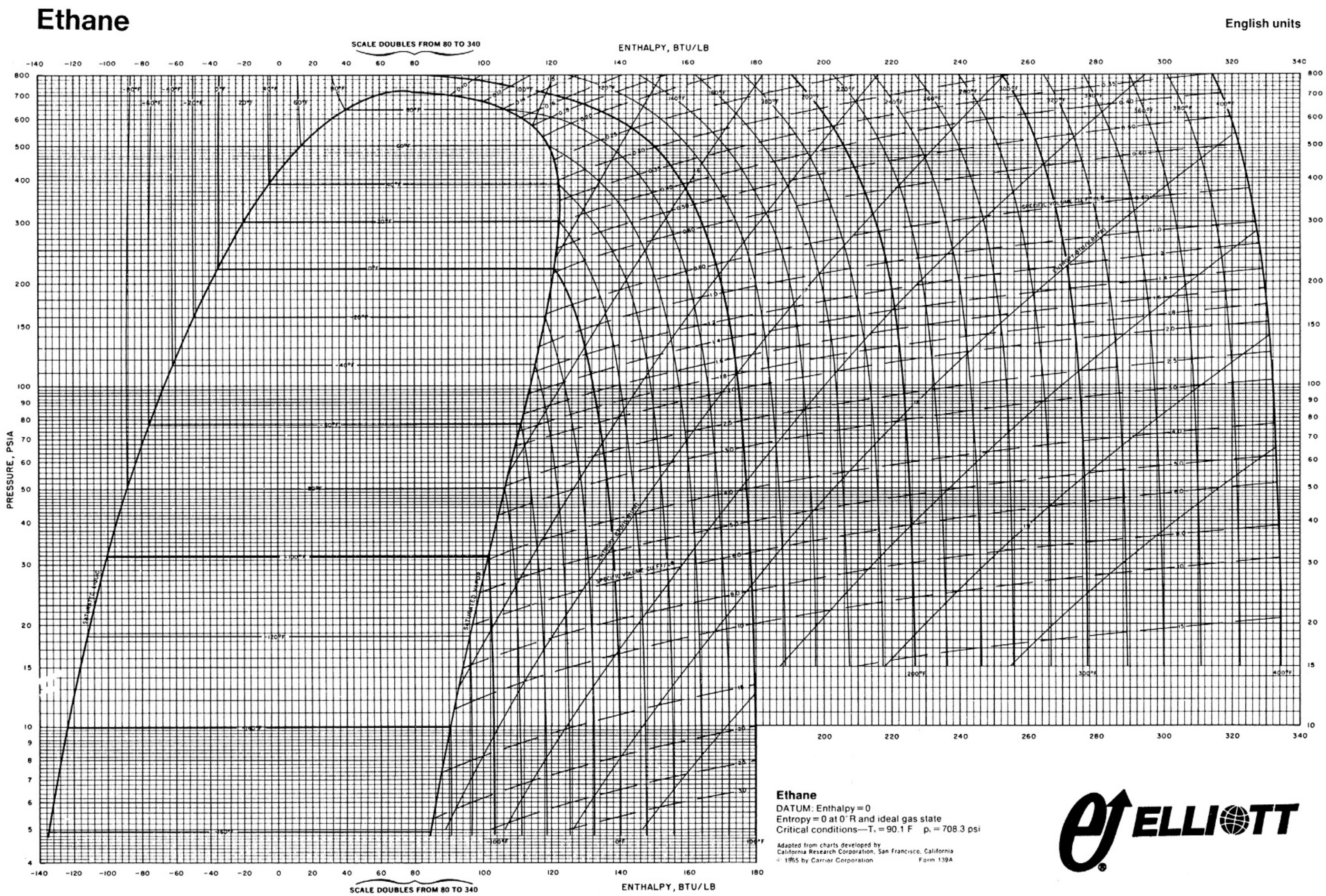


FIG. B.13 Mollier diagram for ethane. *(Used with permission of Elliott Company, Jeannette, PA.)*

Ethane

Metric units

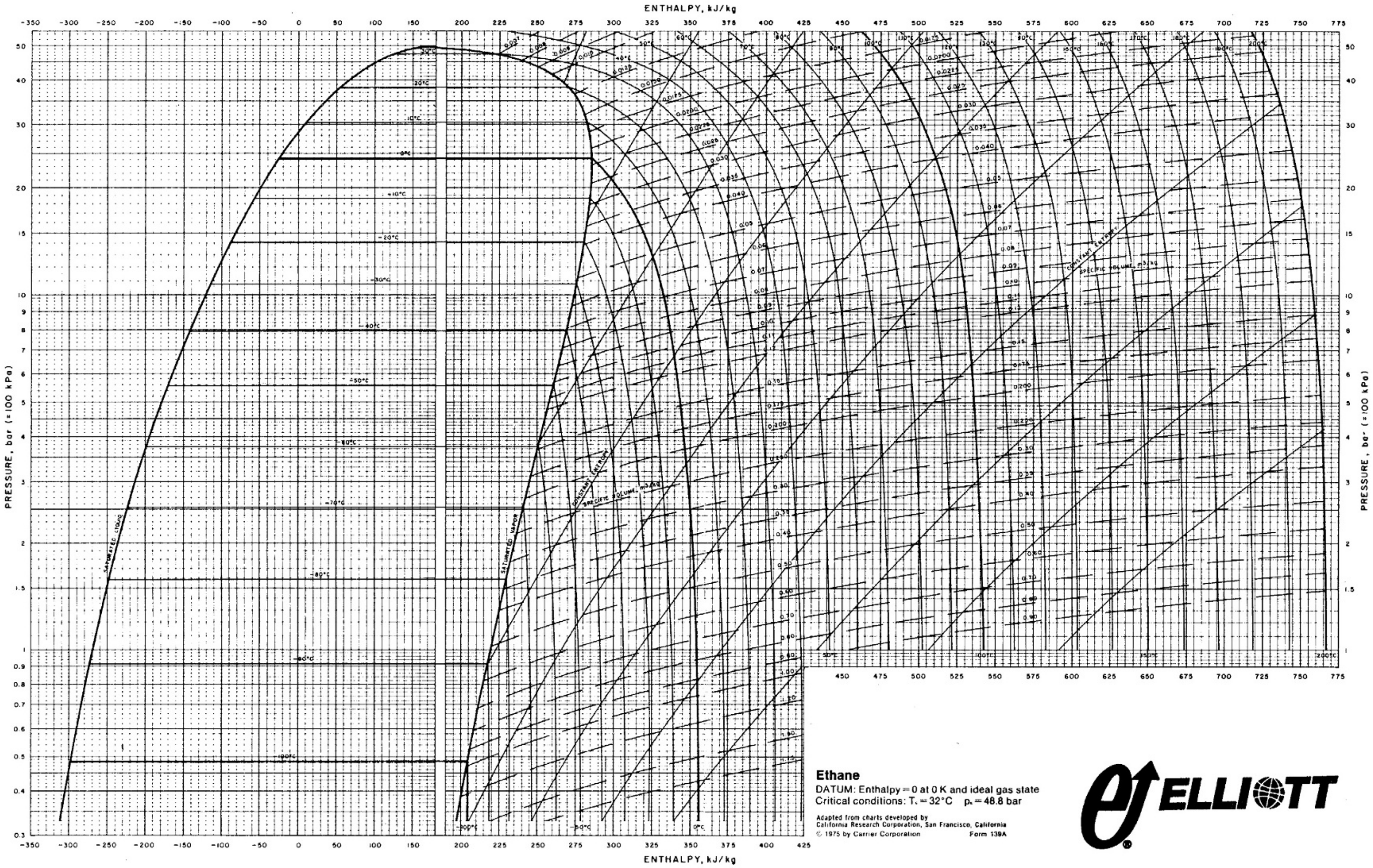


FIG. B.14 Mollier diagram for ethane. *(Used with permission of Elliott Company, Jeannette, PA.)*

Ethylene

English units

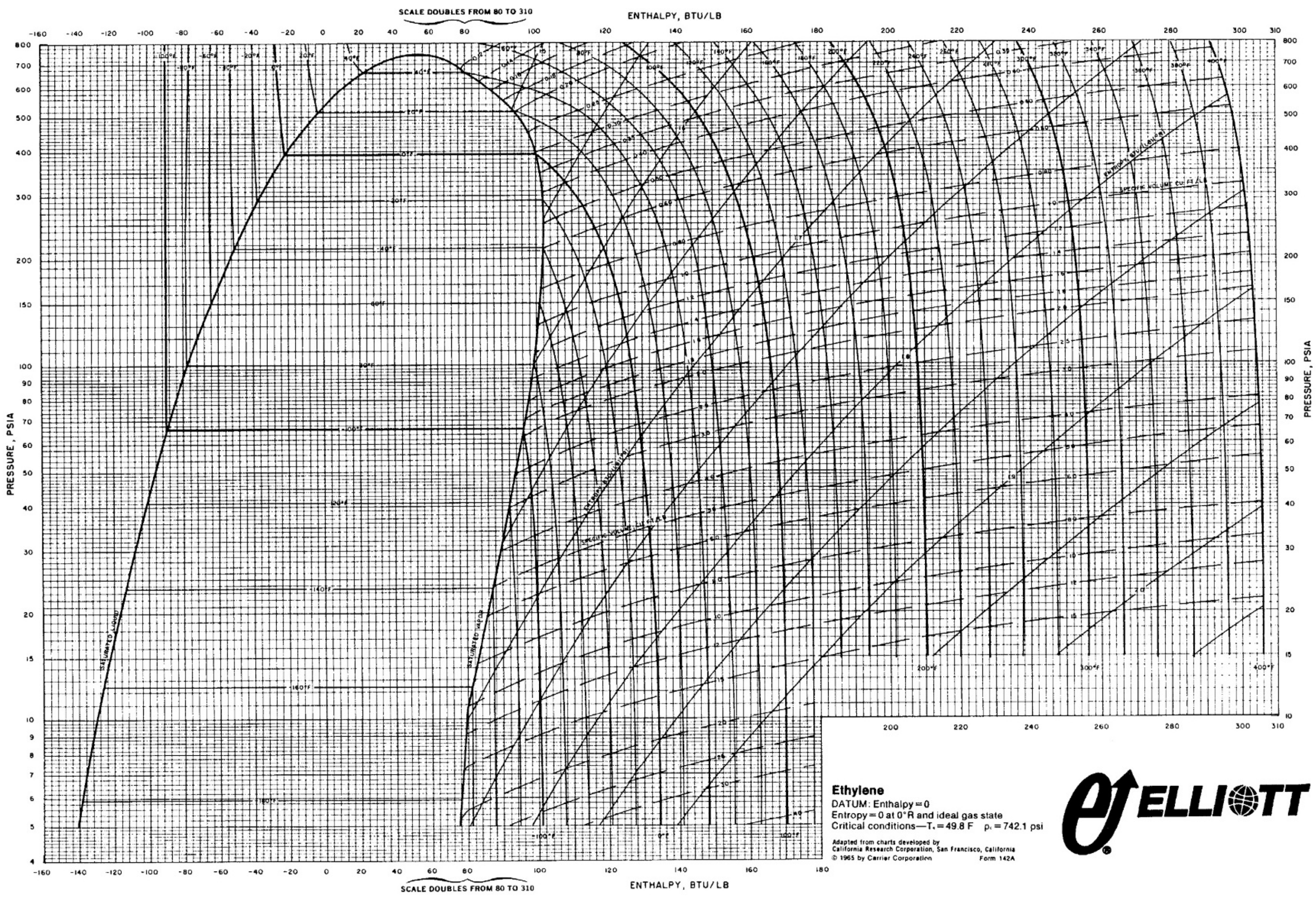


FIG. B.15 Mollier diagram for ethylene. *(Used with permission of Elliott Company, Jeannette, PA.)*

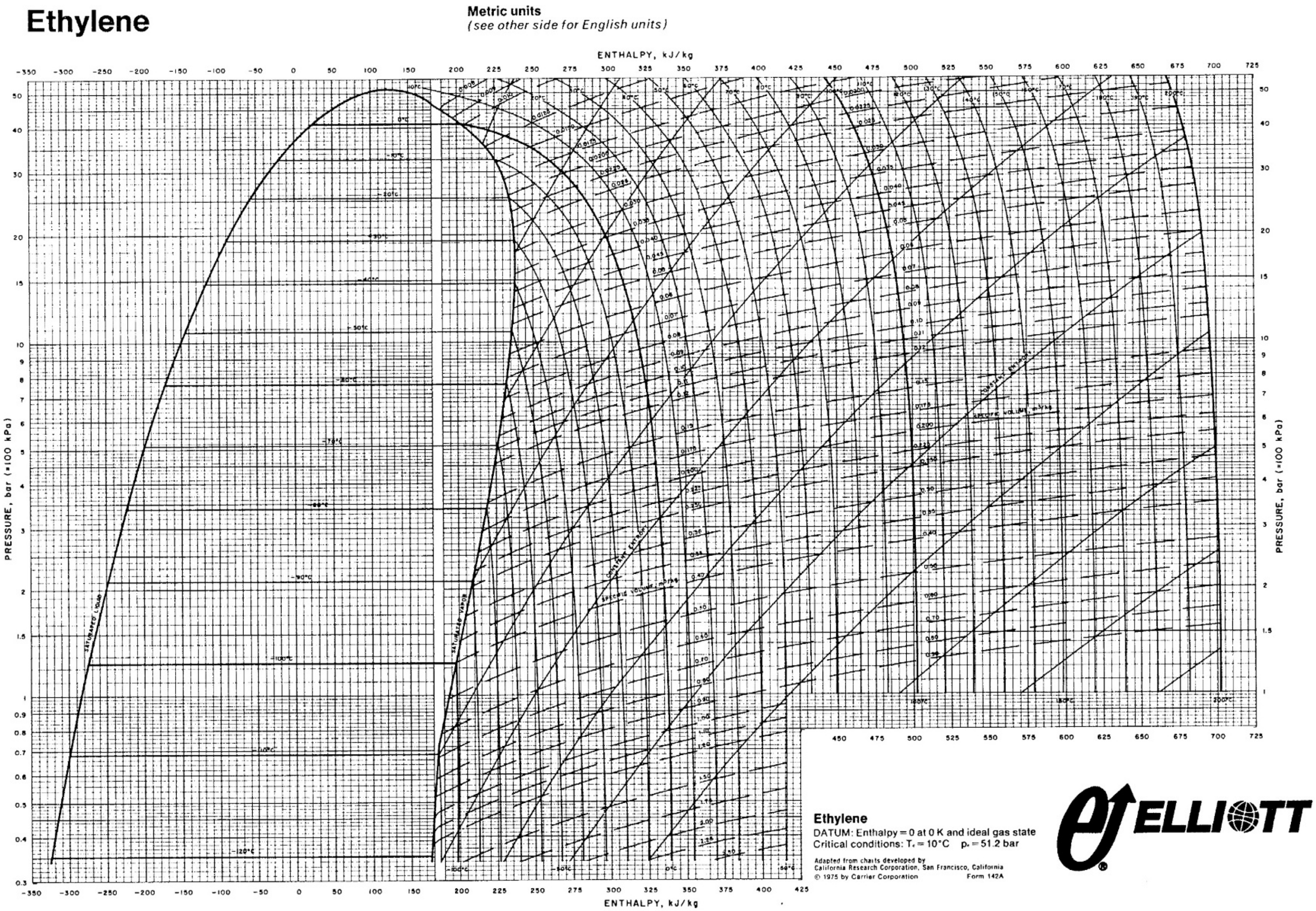


FIG. B.16 Mollier diagram for ethylene. *(Used with permission of Elliott Company, Jeannette, PA.)*

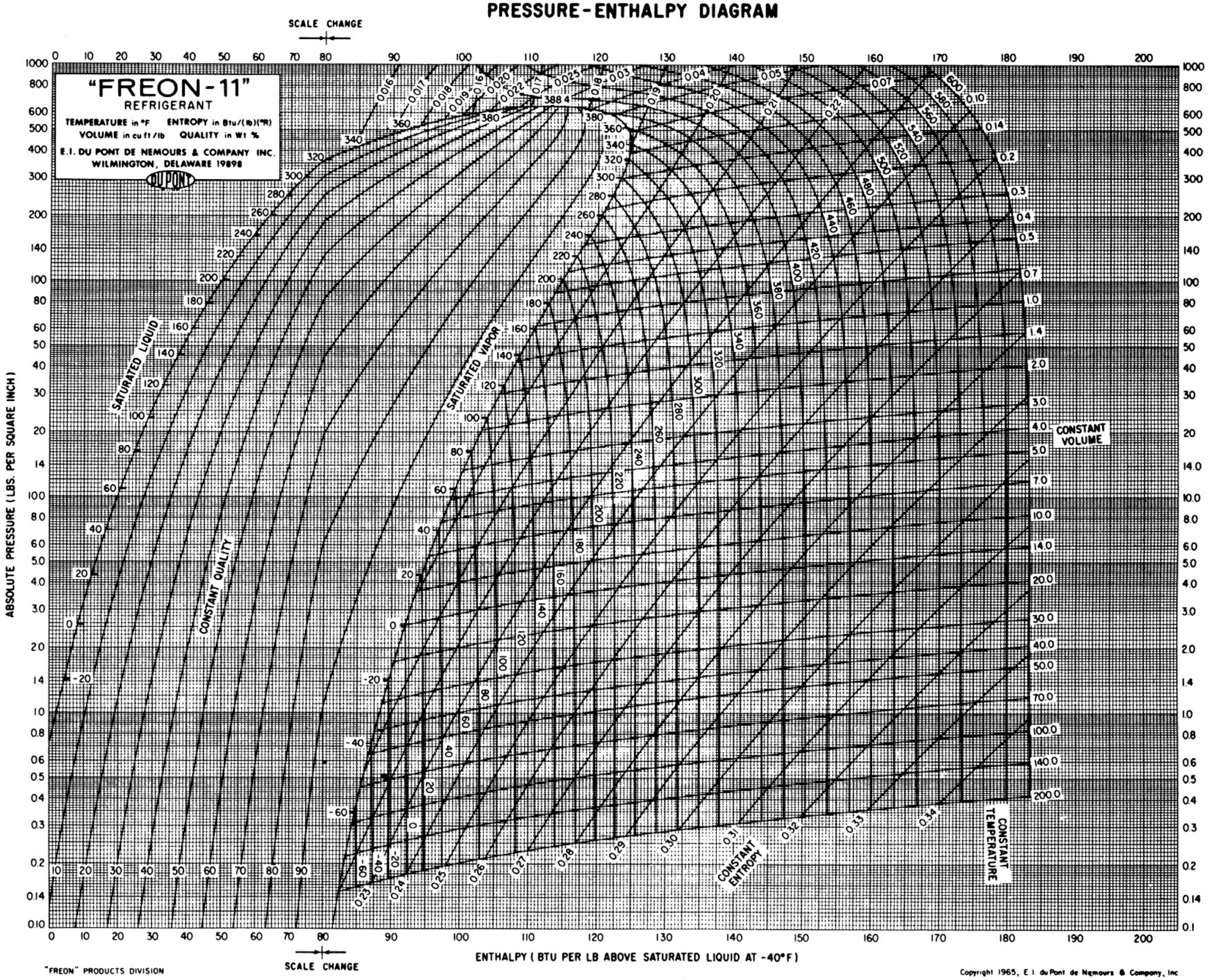


FIG. B.17 Pressure-enthalpy diagram for Freon-11. *(From reference Engineering data book. Tulsa, OK: Natural Gas Processors Suppliers Association; 1967, 1972 with permission of E.I. du Pont de Nemours & Company. Copyright 1965.)*

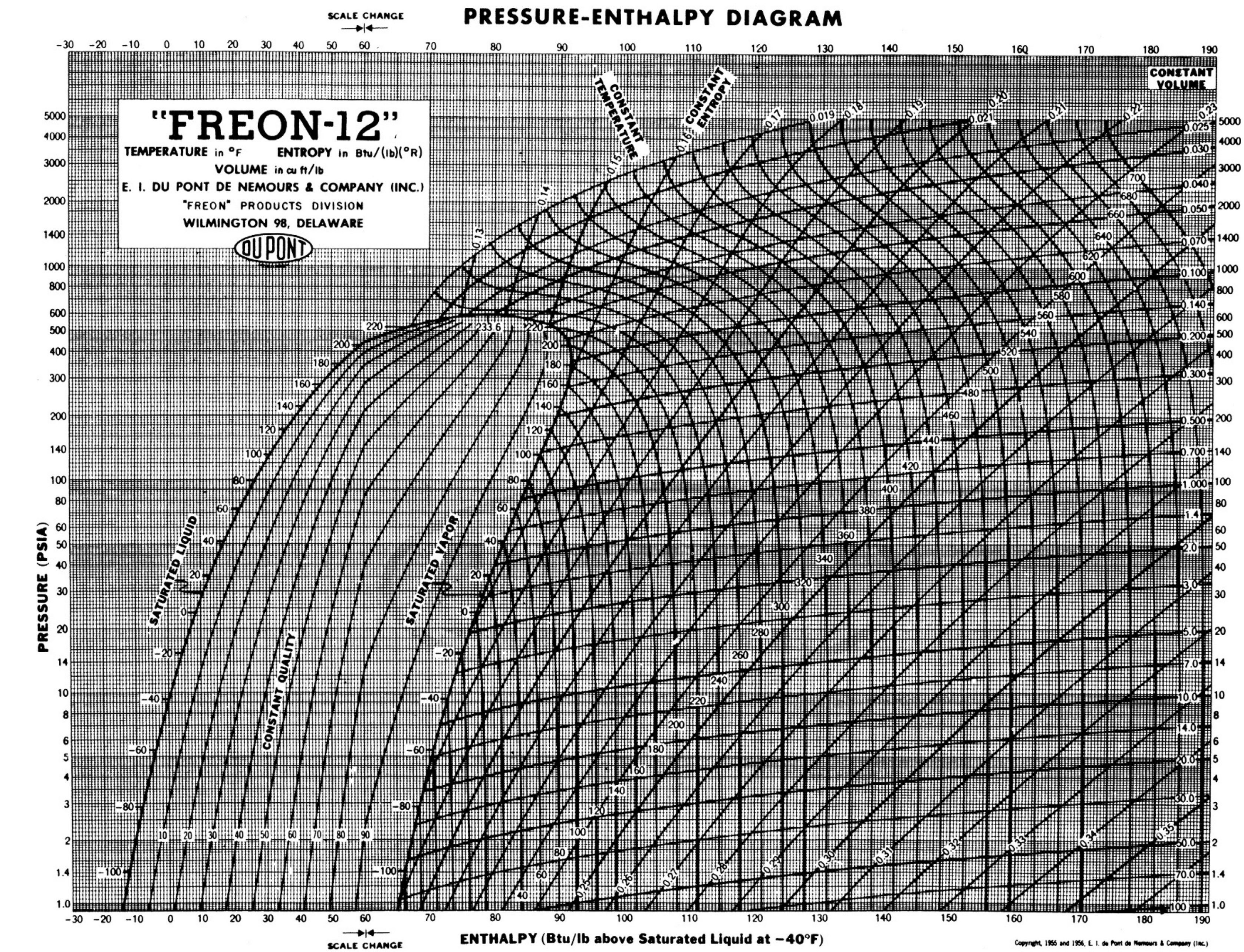
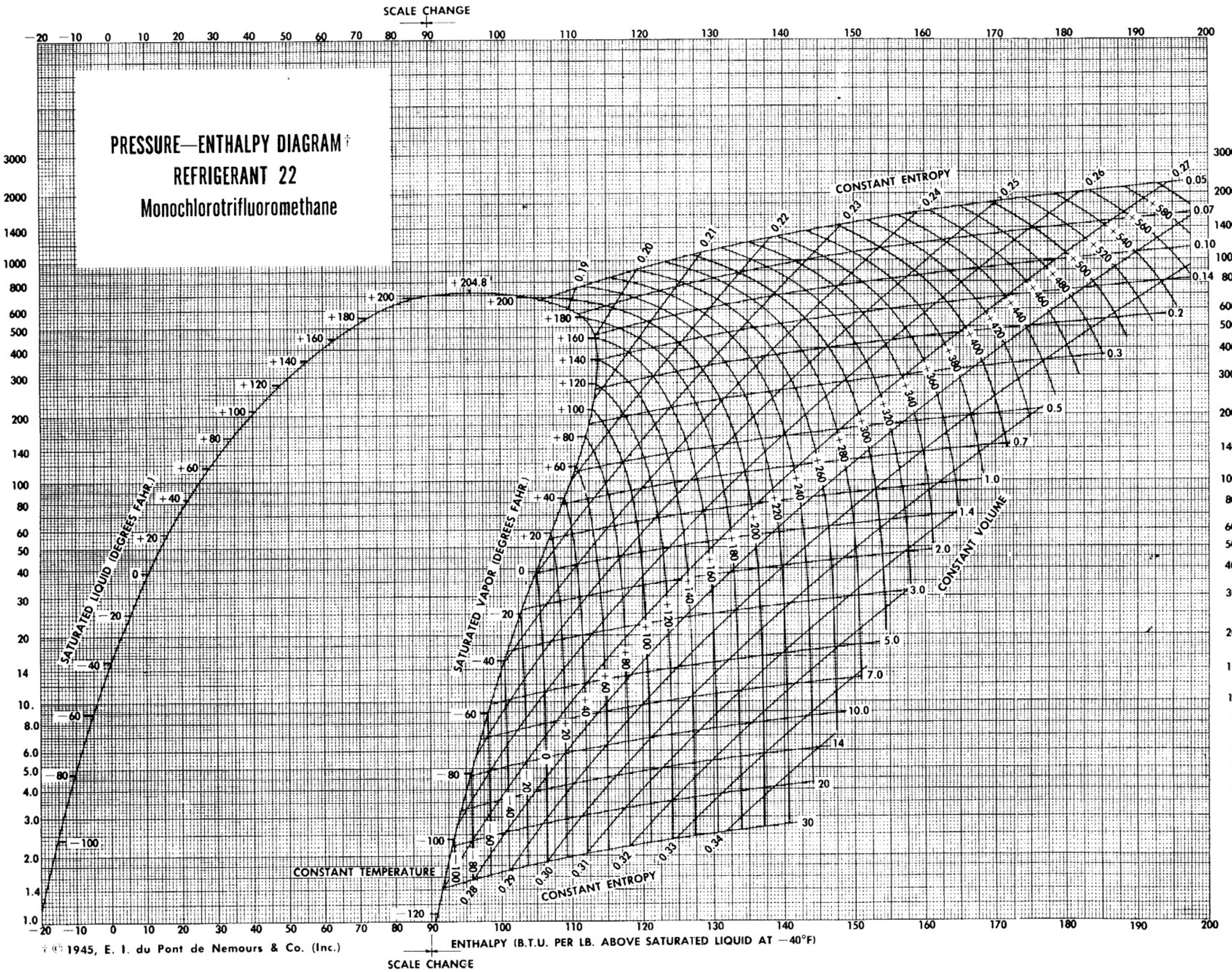


FIG. B.18 Pressure-enthalpy diagram for Freon-12. *(From reference Engineering data book. Tulsa, OK: Natural Gas Processors Suppliers Association; 1967, 1972 with permission of E.I. du Pont de Nemours & Company. Copyright 1965.)*



7

FIG. B.19 Pressure-enthalpy diagram for refrigerant 22. *(From reference Properties of commonly used refrigerants. Washington, D.C.: Air Conditioning and Refrigeration Institute; 1957, with permission of E.I. du Pont de Nemours & Company. Copyright 1945.)*

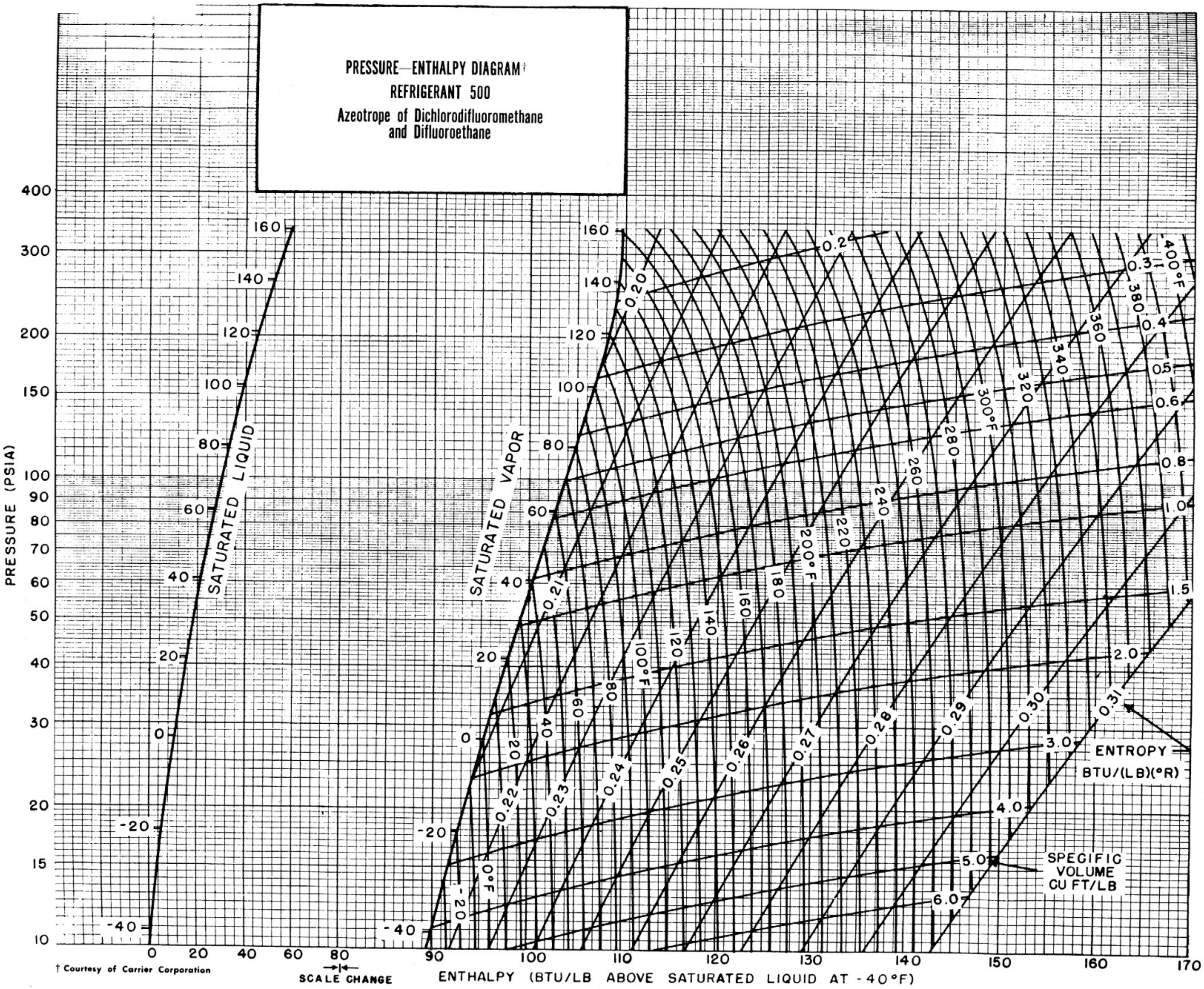


FIG. B.20 Pressure-enthalpy diagram for refrigerant 500. *(From reference Properties of commonly used refrigerants. Washington, D.C.: Air Conditioning and Refrigeration Institute; 1957, courtesy of Carrier Corporation.)*

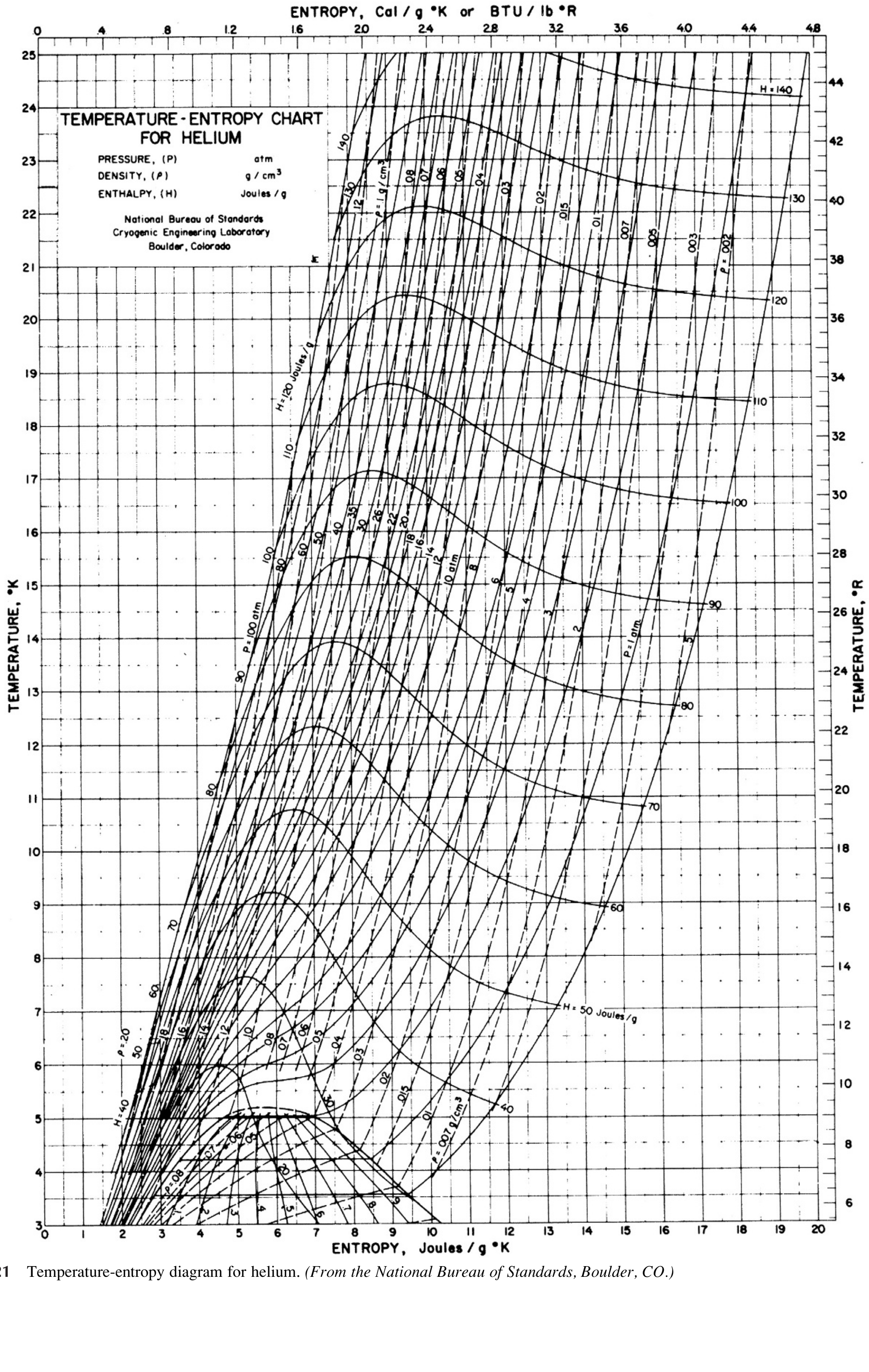


FIG. B.21 Temperature-entropy diagram for helium. *(From the National Bureau of Standards, Boulder, CO.)*

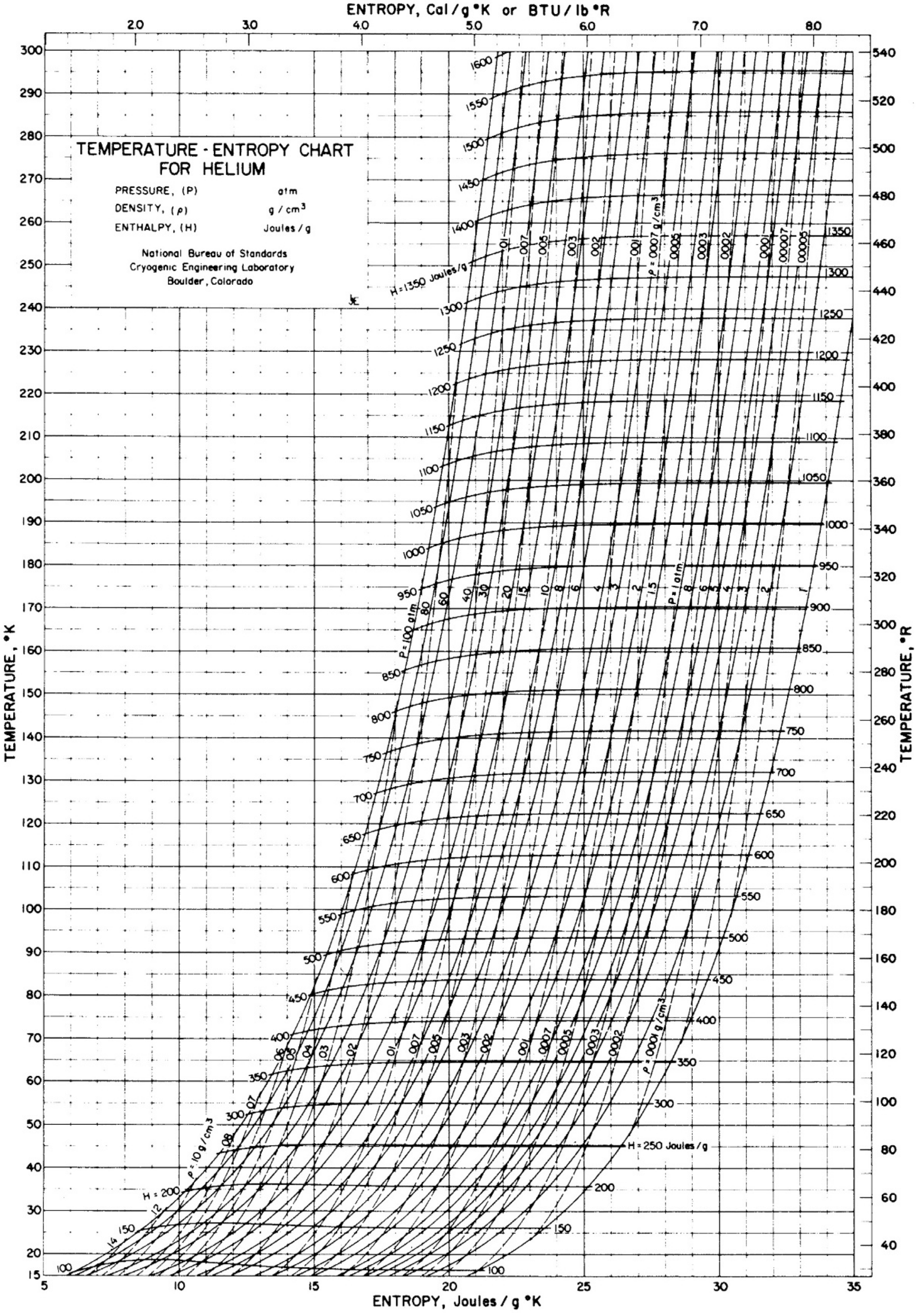


FIG. B.21, CONT'D

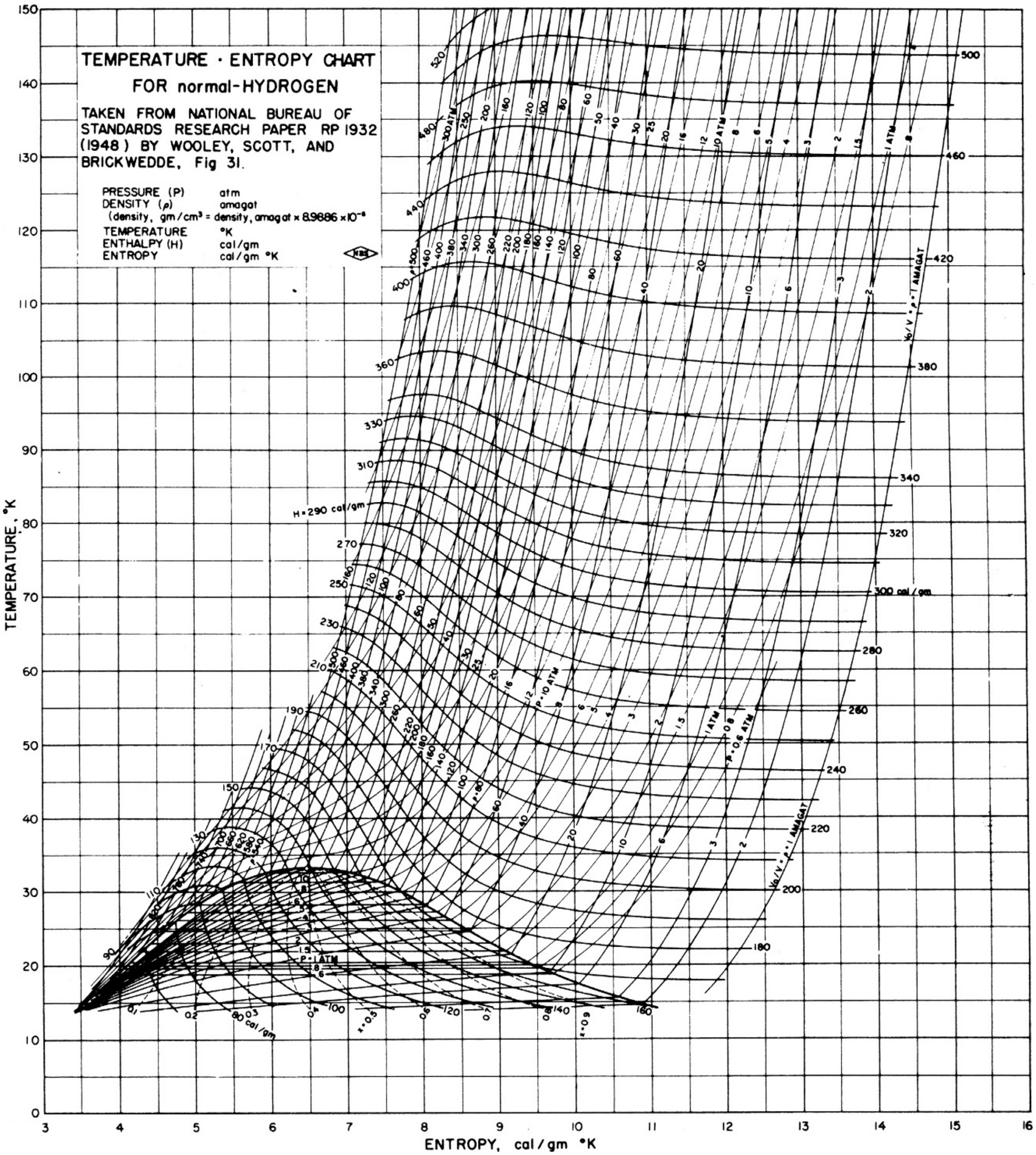


FIG. B.22 Temperature-entropy diagram for hydrogen. *(From the National Bureau of Standards, Boulder, CO.)*

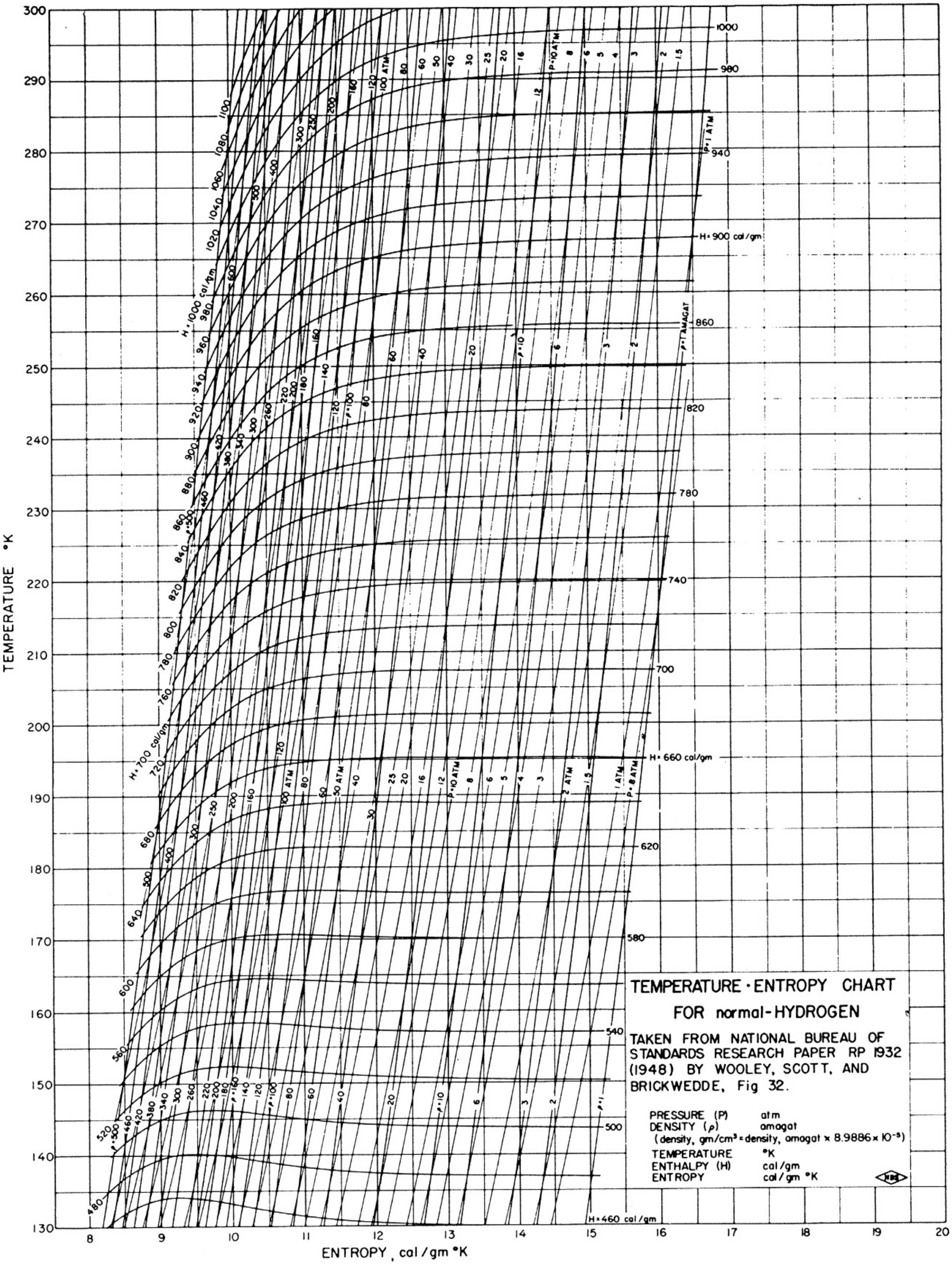


FIG. B.22, CONT'D

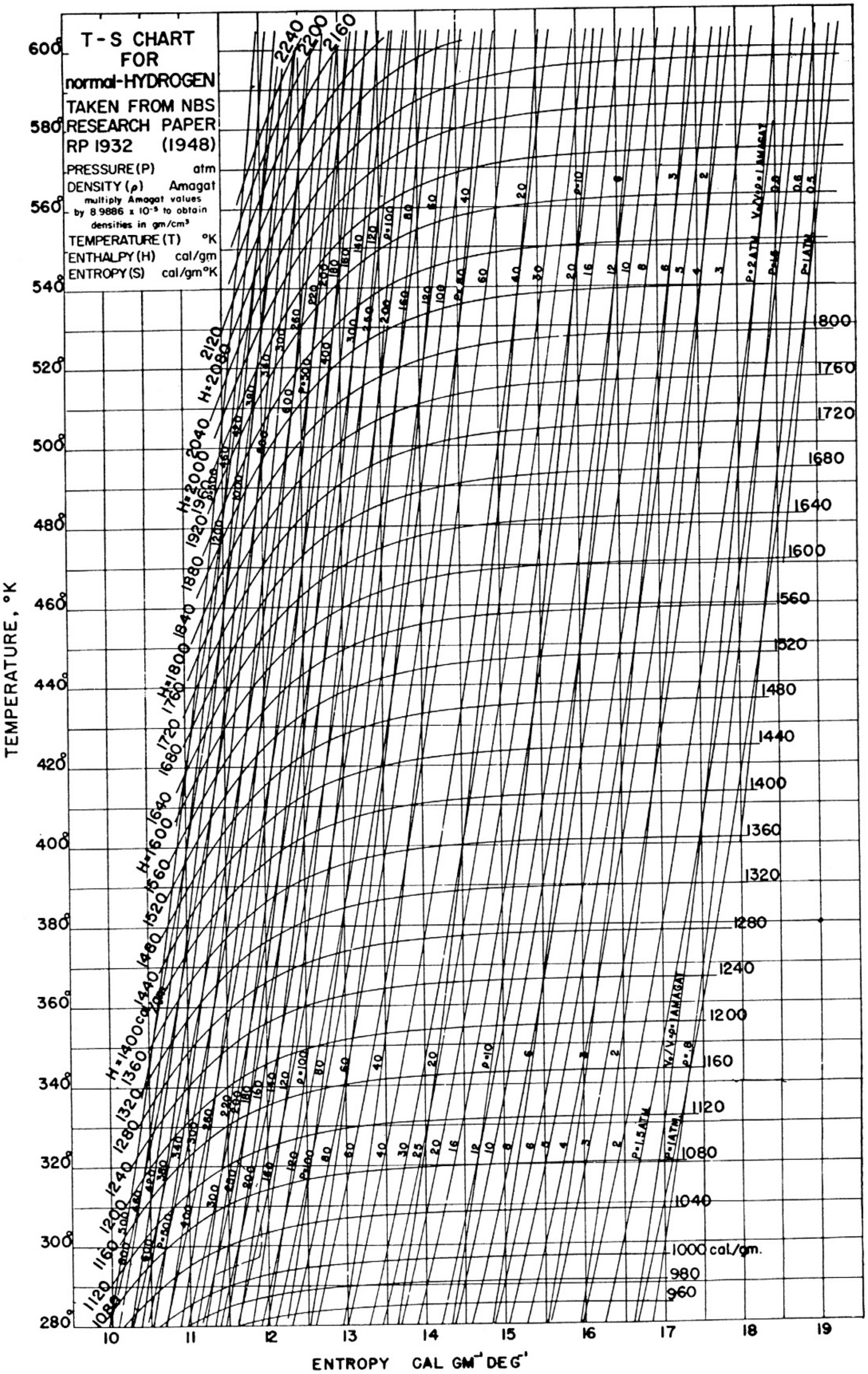


FIG. B.22, CONT'D

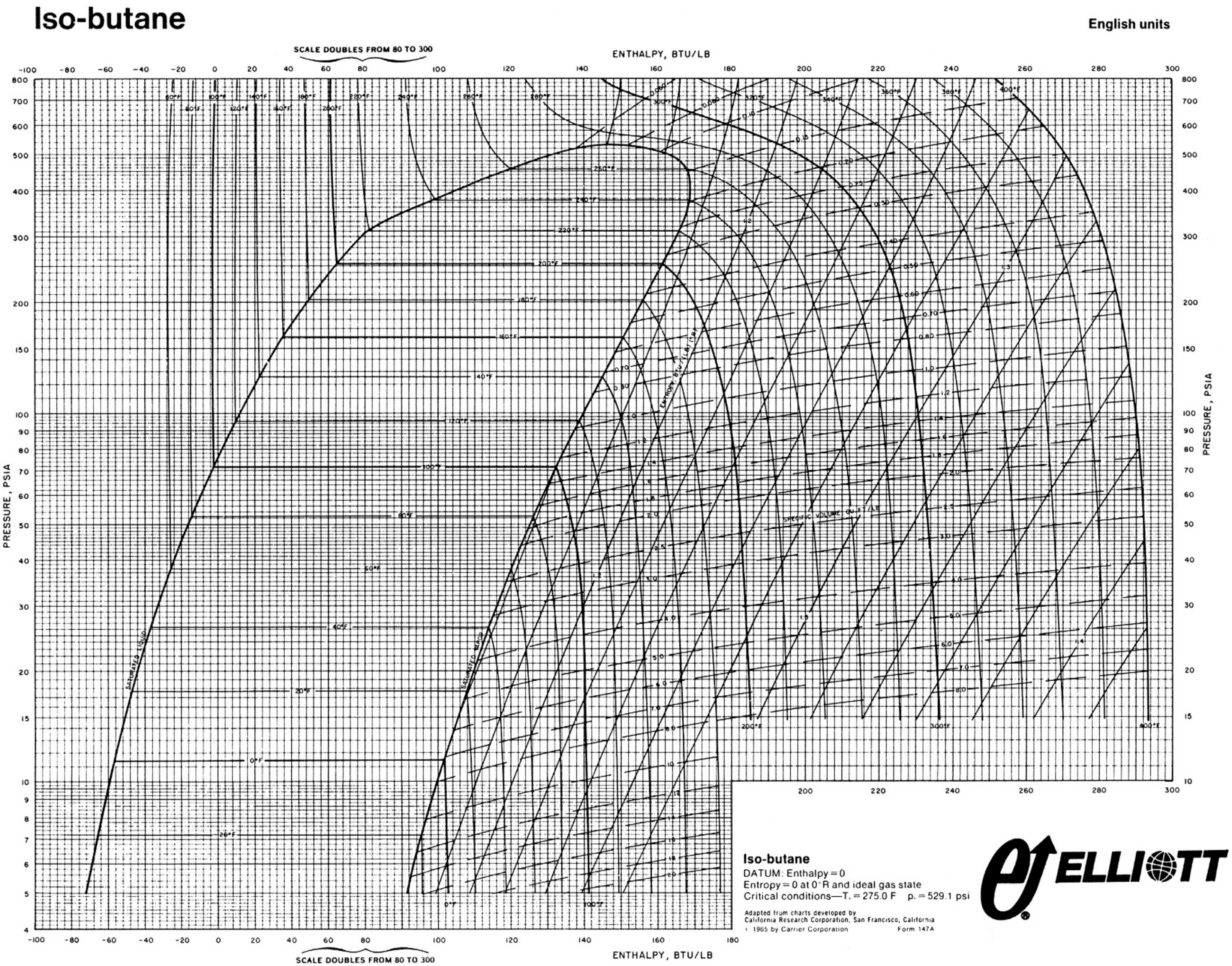


FIG. B.23 Mollier diagram for iso-butane. *(Used with permission of Elliott Company, Jeannette, PA.)*

Iso-butane

Metric units

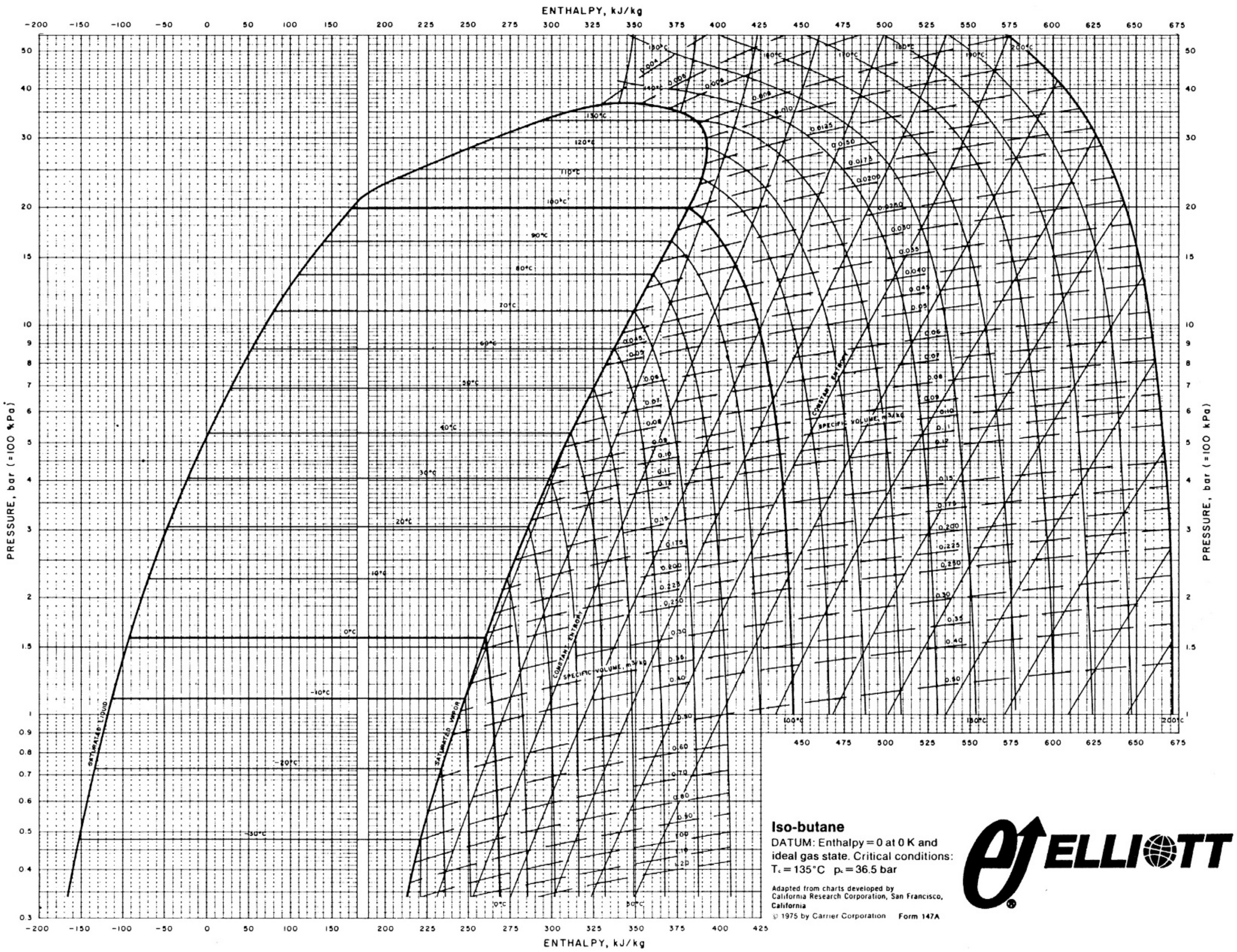


FIG. B.24 Mollier diagram for iso-butane. *(Used with permission of Elliott Company, Jeannette, PA.)*

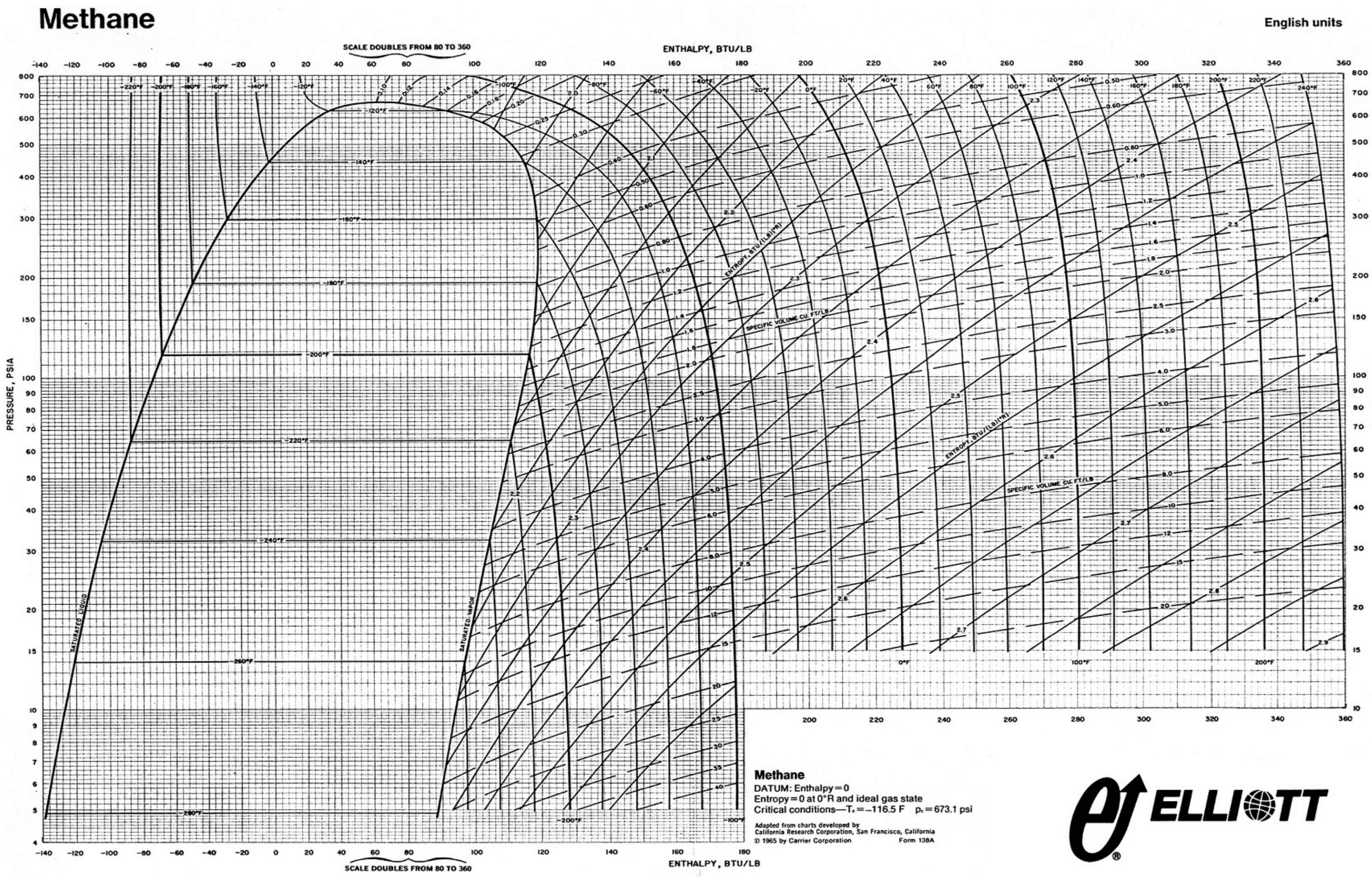


FIG. B.25 Mollier diagram for methane. *(Used with permission of Elliott Company, Jeannette, PA.)*

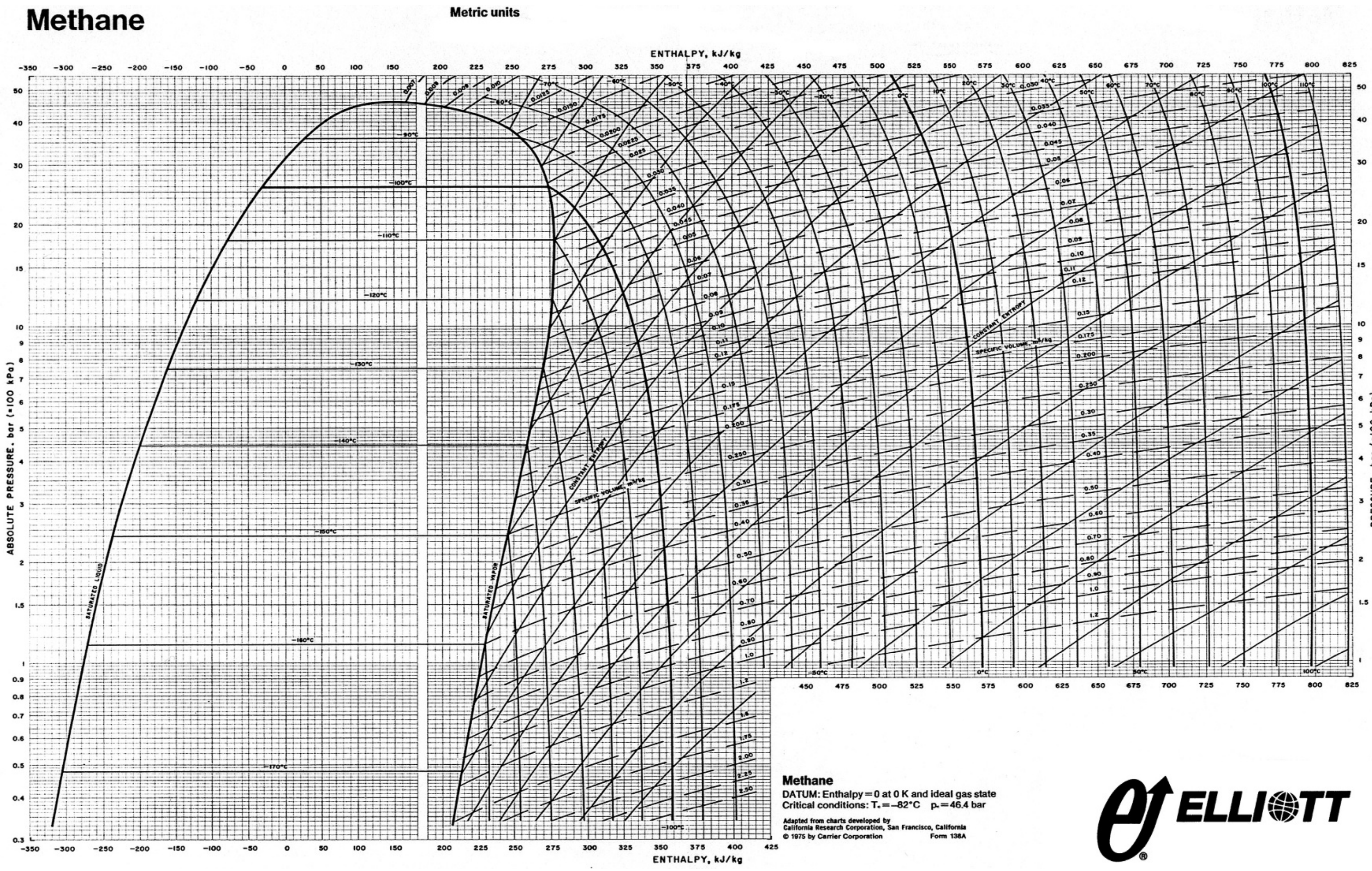


FIG. B.26 Mollier diagram for methane. *(Used with permission of Elliott Company, Jeannette, PA.)*

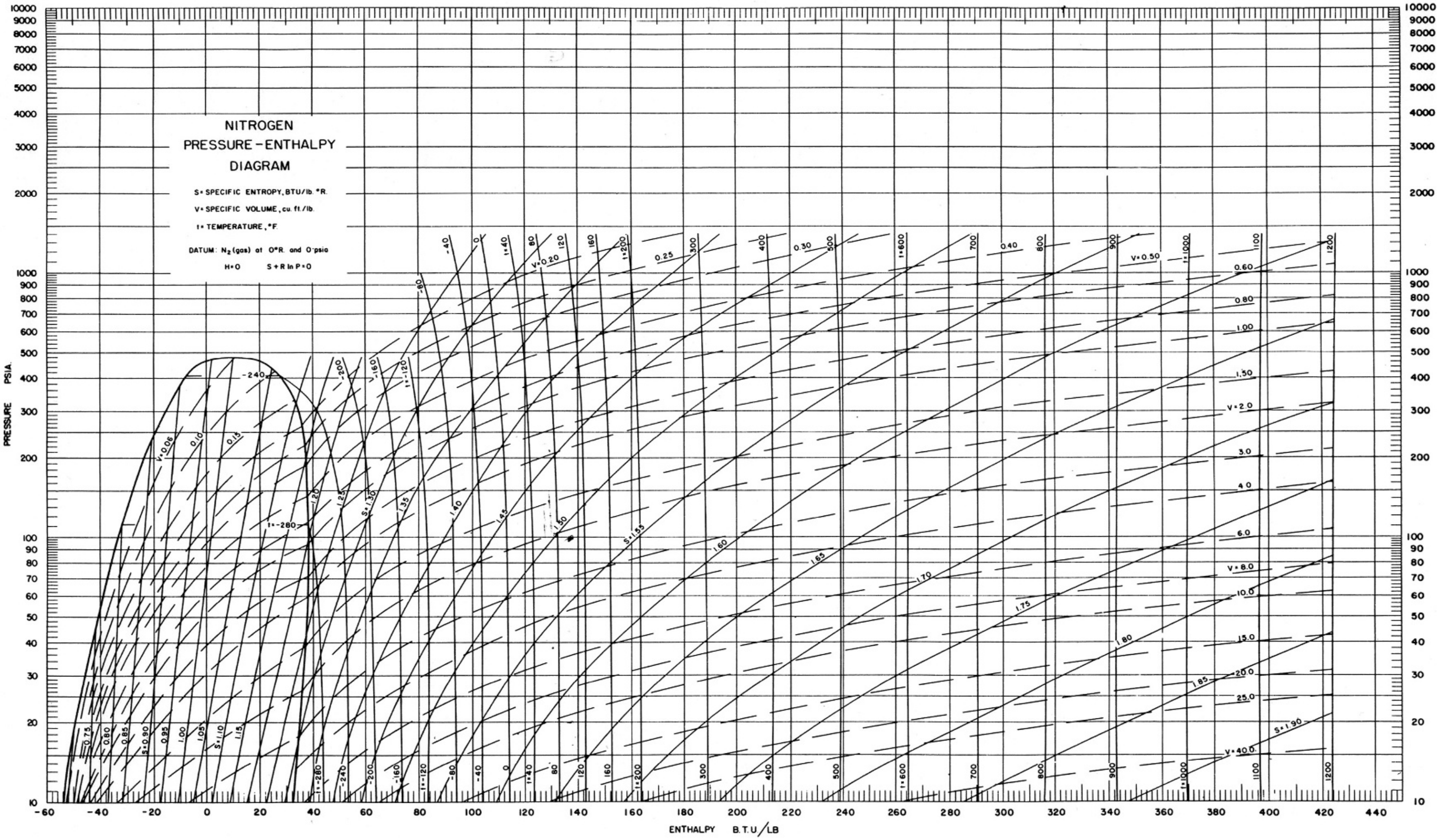


FIG. B.27 Pressure-enthalpy diagram for nitrogen. *(From reference Engineering data book. Tulsa, OK: Natural Gas Processors Suppliers Association; 1967, 1972. Thermo properties of non-hydrocarbons, by Canjar LN, Pollock EK, Cadman TW, Lee WE, Manning FS. Copyright © 1966 by Gulf Publishing Company, Houston, TX. Used with permission. All rights reserved.)*

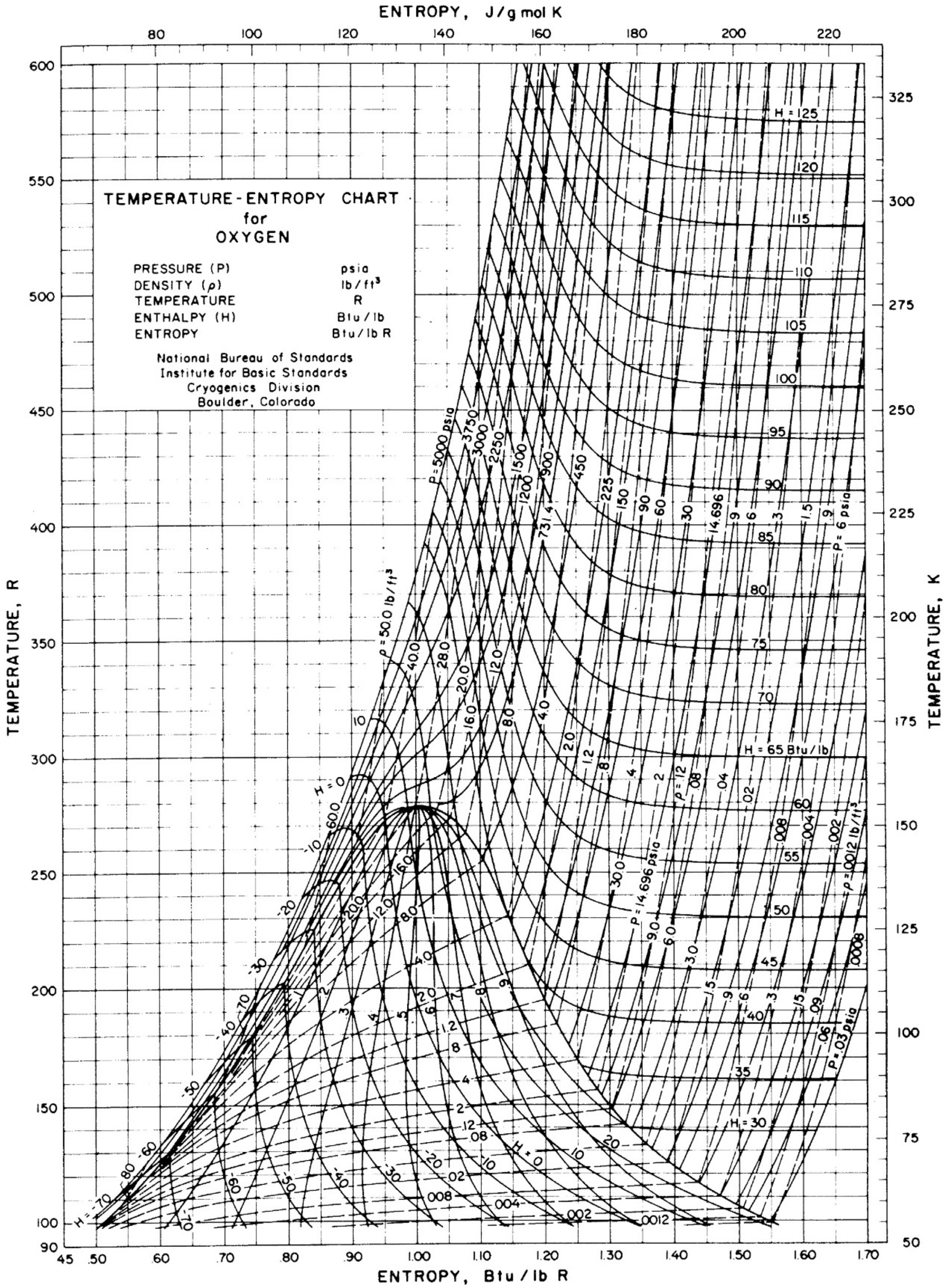


FIG. B.28 Temperature-entropy diagram for oxygen. *(From National Bureau of Standards, Boulder, CO.)*

Propane

English units

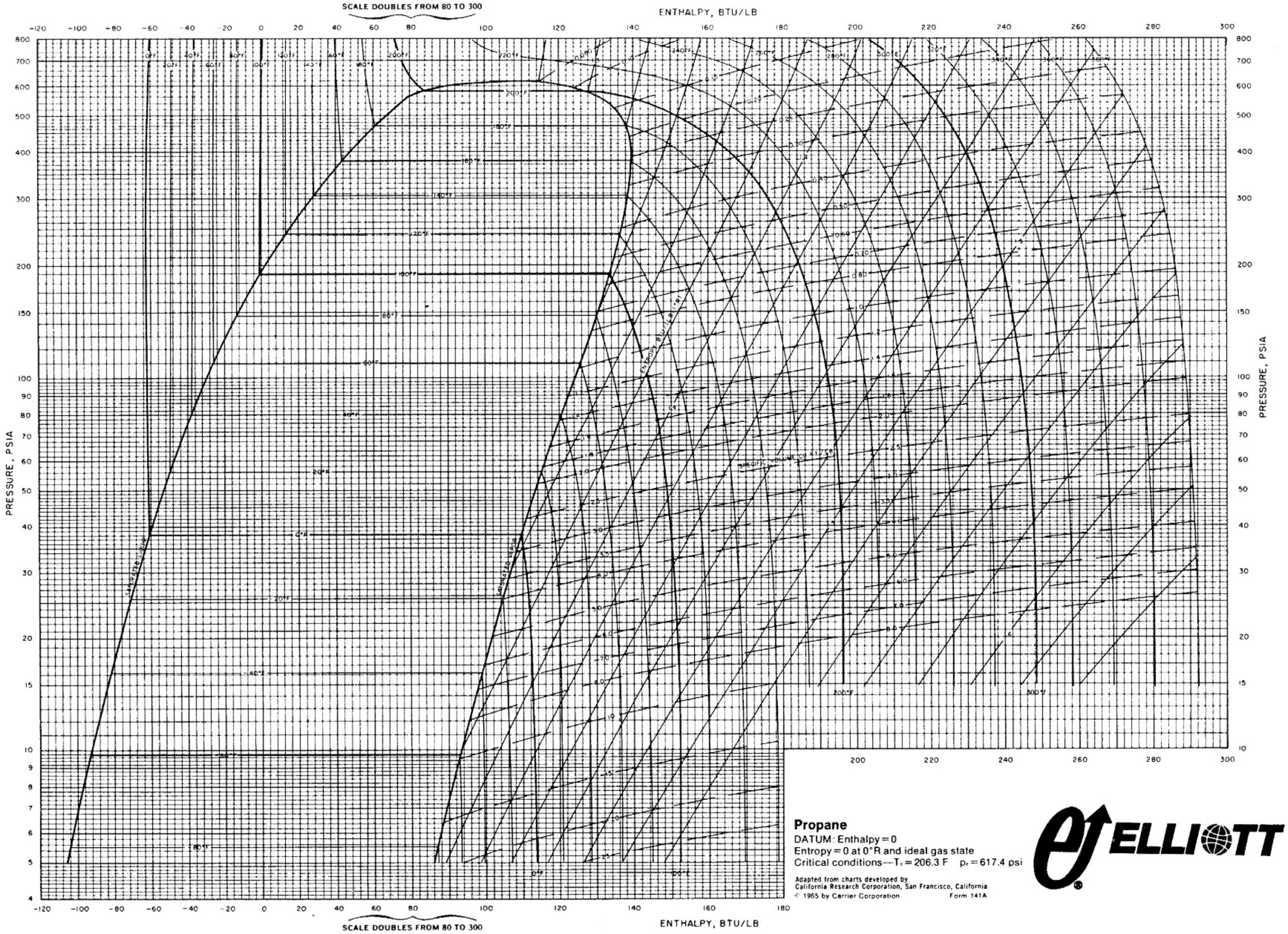


FIG. B.29 Mollier diagram for propane. *(Used with permission of Elliott Company, Jeannette, PA.)*

Propane

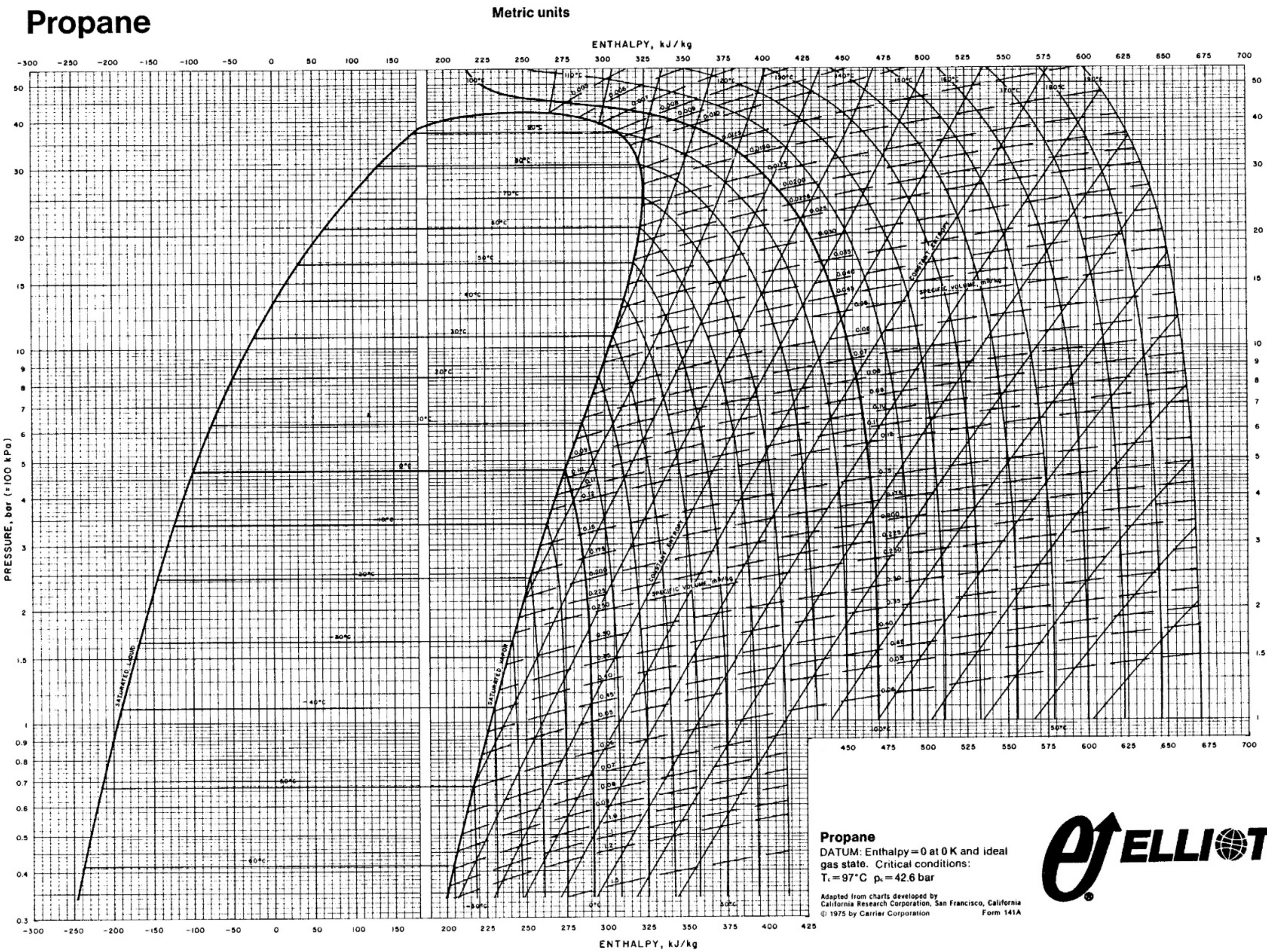


FIG. B.30 Mollier diagram for propane. *(Used with permission of Elliott Company, Jeannette, PA.)*

Propylene

English units

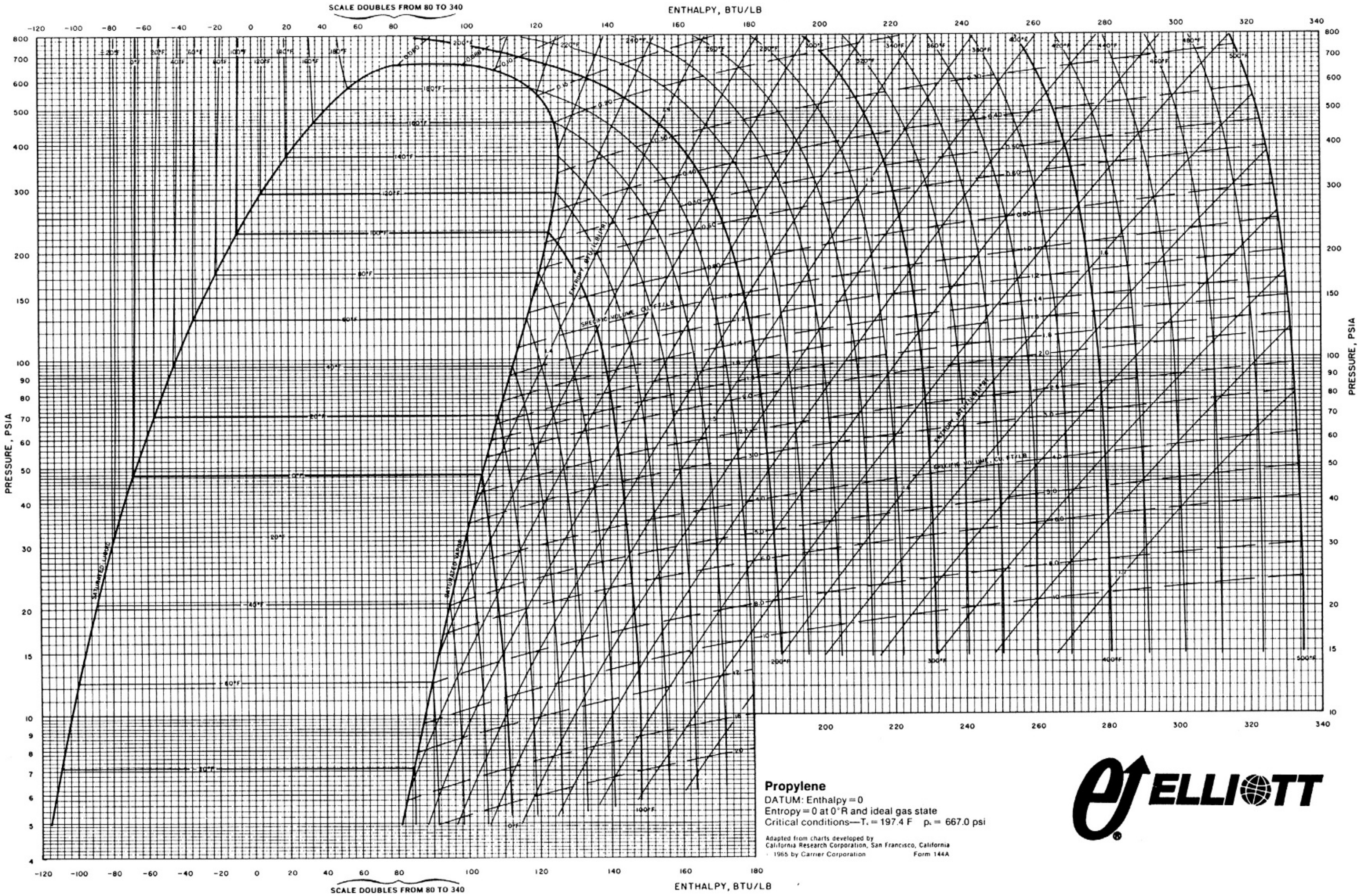


FIG. B.31 Mollier diagram for propylene. *(Used with permission of Elliott Company, Jeannette, PA.)*

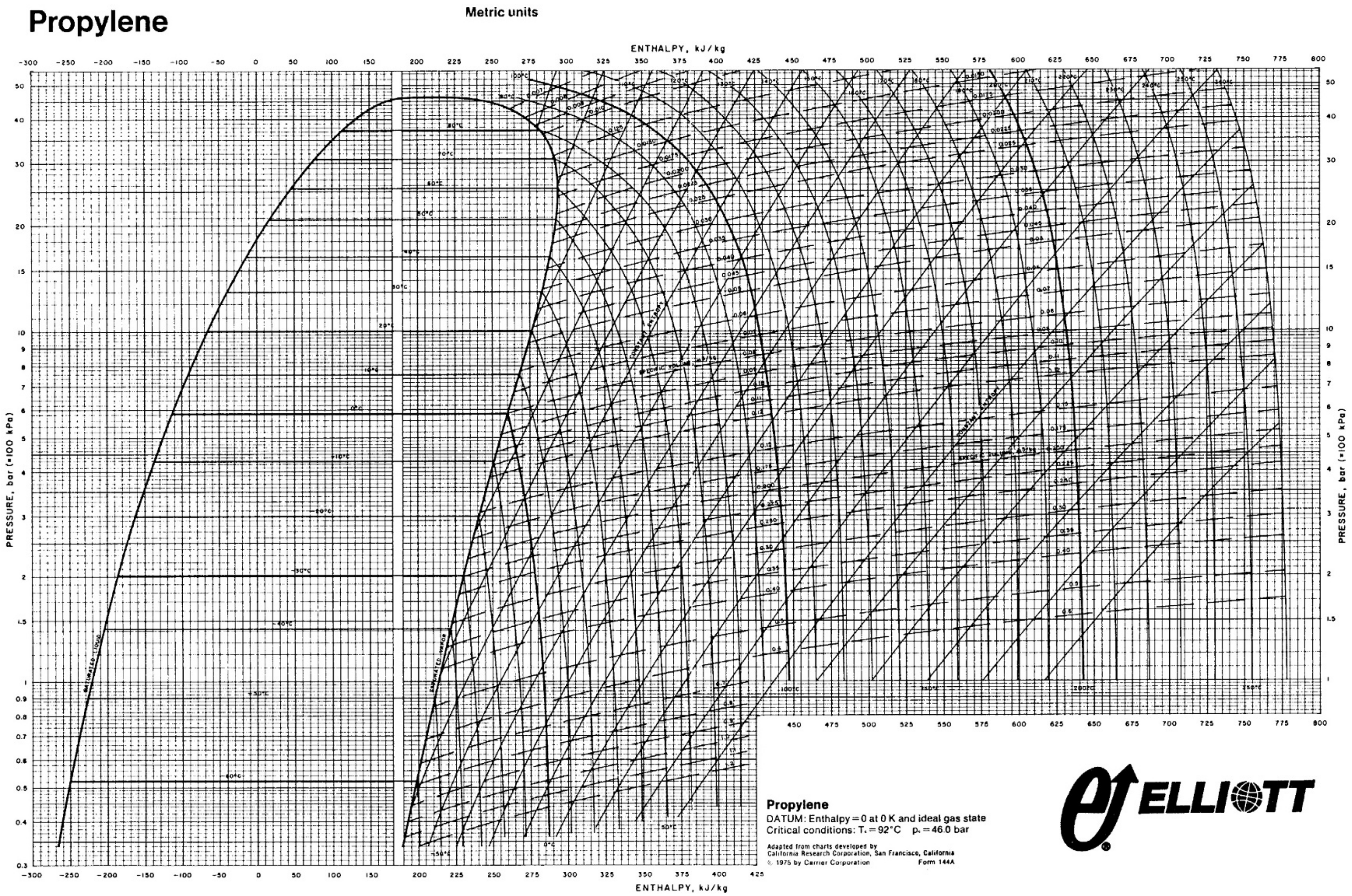


FIG. B.32 Mollier diagram for propylene. *(Used with permission of Elliott Company, Jeannette, PA.)*

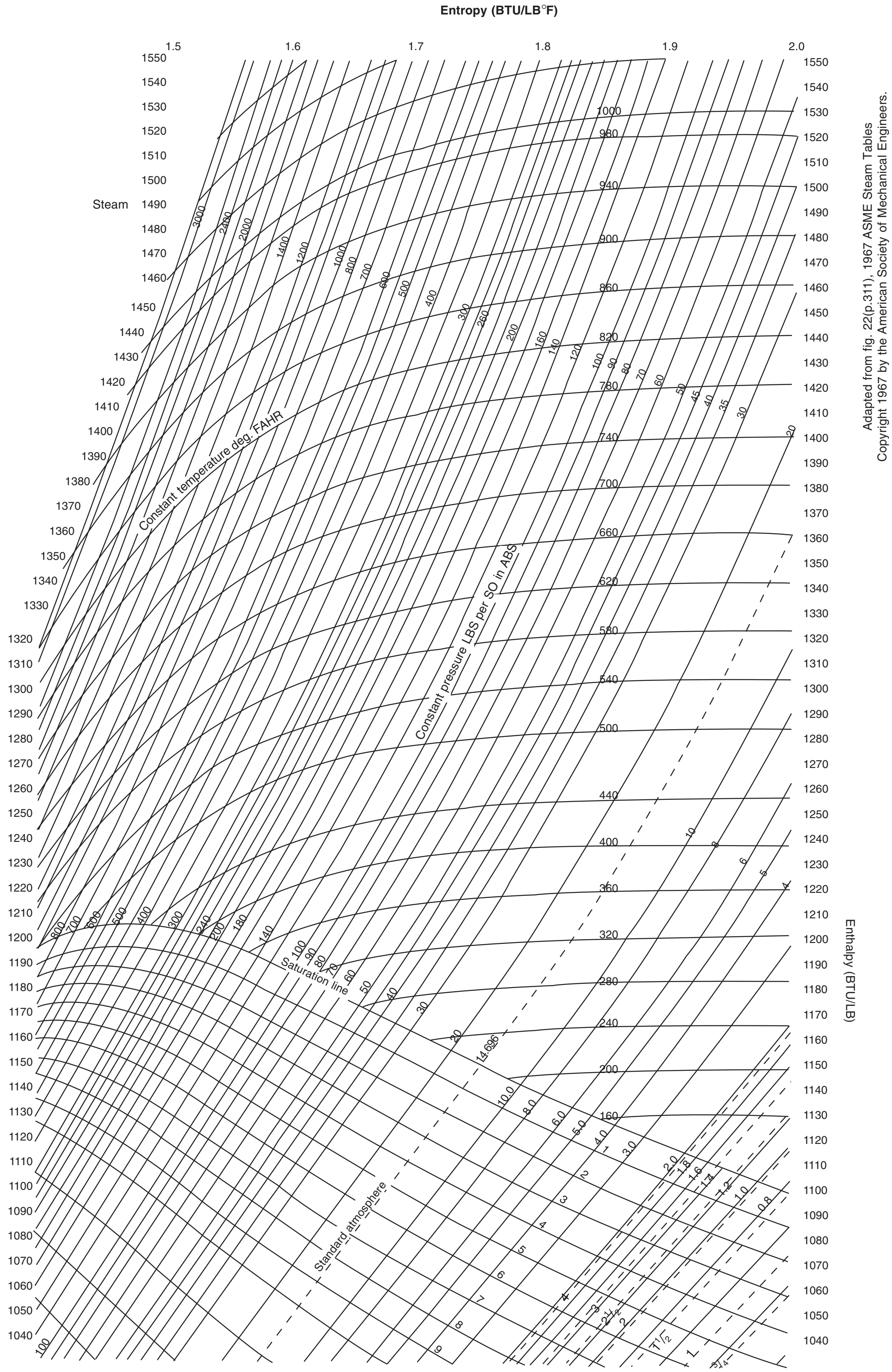

Entropy (BTU/LB°F)
1.5
1.6
1.7
1.8
1.9
2.0
Steam
Constant temperature deg. FAHR
Constant pressure LBS per SQ in ABS
Saturation line
Standard atmosphere
Enthalpy (BTU/LB)
Adapted from fig. 22(p.311), 1967 ASME Steam Tables
Copyright 1967 by the American Society of Mechanical Engineers.

FIG. B.33

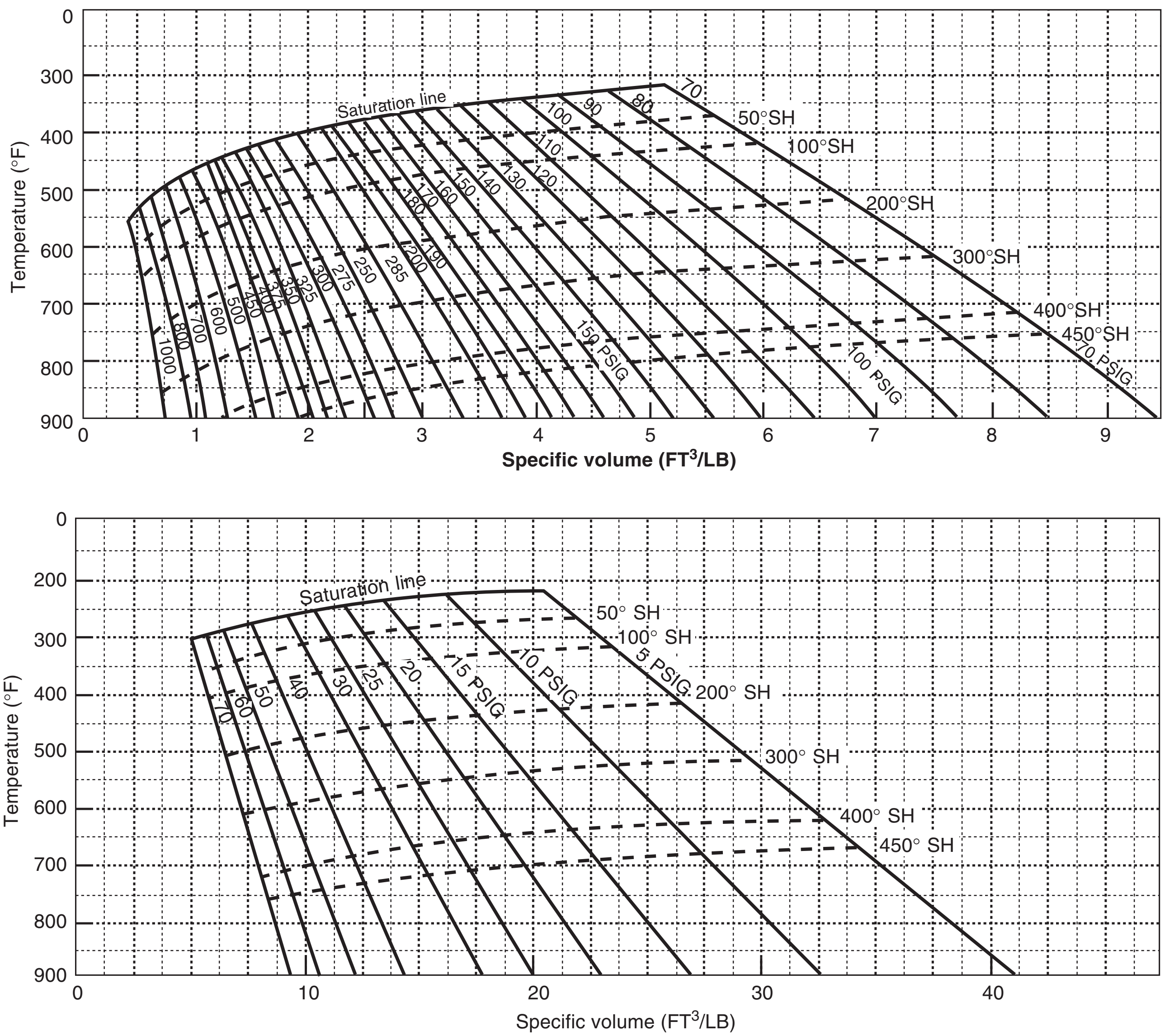


FIG. B.34 Specific volume for steam. *(Used with permission of Elliott Company, Jeannette, PA.)*

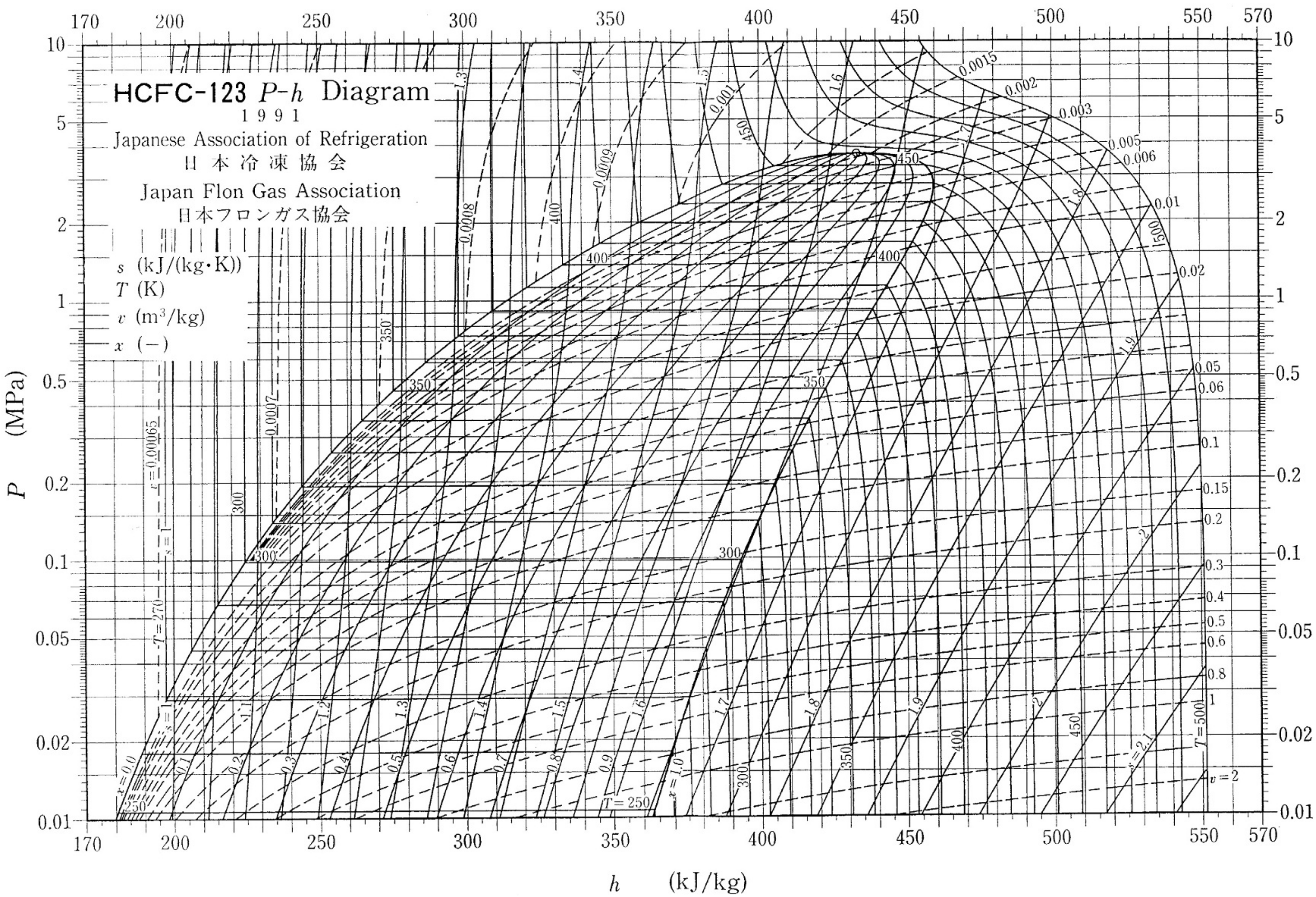


FIG. B.35 Refrigerant 123. *(Courtesy Japanese Association of Refrigeration and Japan Flon Gas Association.)*

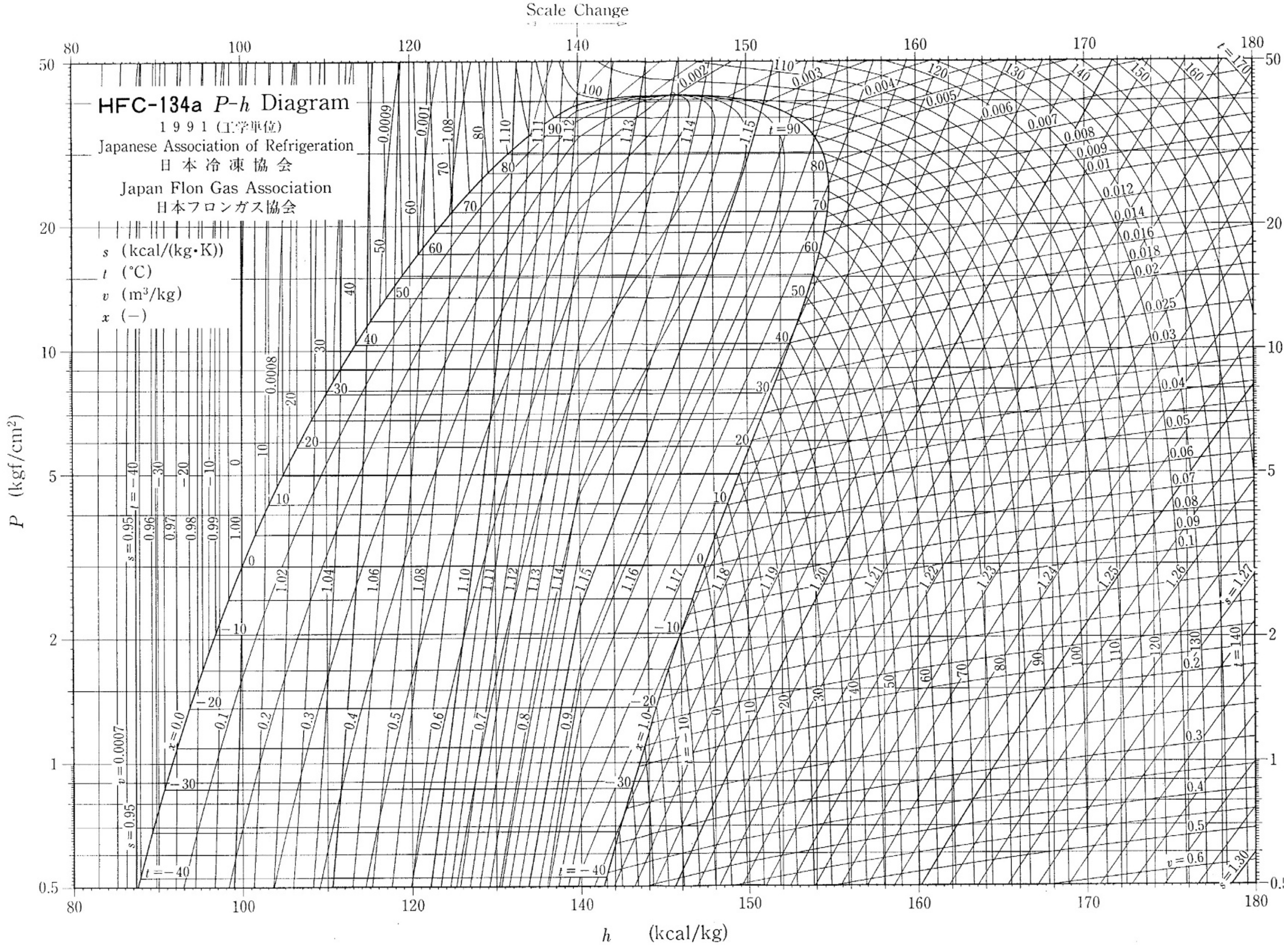


FIG. B.36 Refrigerant 135a. *(Courtesy Japanese Association of Refrigeration and Japan Flon Gas Association.)*

Appendix C

Conversion Tables

A Table of Conversion Factors is Provided in Alphabetical Order		
To Convert	**Multiply By**	**To Obtain**
A		
Acres	4.35×10^4	square feet
Acres	4047	square meters
Acres	0.001562	square miles
Acres	4840	square yards
Ampere-hours	3600	coulombs
Ampere-hours	0.03731	faradays
Ampere-turns	1.257	gilberts
Angstrom unit	3.937×10^{-9}	inches
Angstrom unit	1×10^{-10}	meters
Angstrom unit	1×10^{-4}	microns or (μ)
Ares	0.02471	acres (U.S.)
Ares	119.6	sq. yards
Ares	100	sq. meters
Astronomical unit	1.495×10^8	kilometers
Atmospheres	0.007348	tons/in^2
Atmospheres	1.058	tons/ft^2
Atmospheres	76	cm of mercury (at 0°C)
Atmospheres	33.9	feet of water (at 4°C)
Atmospheres	29.92	inches of mercury (at 0°C)
Atmospheres	0.76	meters of mercury (at 0°C)
Atmospheres	760	millimeters of mercury (at 0°C)
Atmospheres	1.03323	kg/cm^2
Atmospheres	1.0333×10^4	kg/m^2
Atmospheres	14.7	lb/in^2
Atmospheres	1.01325	bar

Continued

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd

To Convert	Multiply By	To Obtain
B		
Barrels (U.S., dry)	3.281	bushels
Barrels (U.S., dry)	7056	cu. inches
Barrels (U.S., dry)	105	quarts (dry)
Barrels (U.S., liquid)	31.5	gallons
Barrels (oil)	42	gallons (oil)
Bars	0.986923	atmospheres
Bars	1×10^6	dynes/cm^2
Bars	1.020×10^4	kg/m^2
Bars	2089	lb/ft^2
Bars	14.5038	lb/in^2
Bars	1.01972	kg/cm^2
Bars	750.062	millimeters of mercury (at 0°C)
Bars	29.5300	inches of mercury (at 32°F)
Bars	33.488	ft water (at 60°F)
Barye	1.00	dynes/cm^2
Bolt (U.S., cloth)	36.576	meters
BTU	10.409	liter-atmospheres
BTU	1.0550×10^{10}	ergs
BTU	778.16	foot-pounds
BTU	252	gram-calories
BTU	3.93011×10^{-4}	horsepower-hours
BTU	1055.05	joules
BTU	0.252	kilogram-calories
BTU	107.585	kilogram-meters
BTU	2.928×10^{-4}	kilowatt-hours
BTU/h	0.2162	ft-lb/s
BTU/h	0.070	g-cal/s
BTU/h	3.929×10^{-4}	horsepower
BTU/h	0.2931	watts
BTU/min	12.96	ft-lb/s
BTU/min	0.02356	horsepower
BTU/min	0.01757	kilowatts
BTU/min	17.57	watts
BTU/lb$_m$	2.326	kilojoules/kilogram
BTU/lb$_m$	2.326	kilonewton-meters/kilograms
BTU/ft^2/min	0.0122	watts/in^2
Bucket (Br. dry)	1.8184×10^4	cubic cm
Bushels	1.2445	cubic ft
Bushels	2150.4	cubic in

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd

To Convert	Multiply By	To Obtain
Bushels	0.03524	cubic meters
Bushels	34.24	liters
Bushels	4.0	pecks
Bushels	64	pints (dry)
Bushels	32	quarts (dry)
C		
Calories, gram (mean)	0.0039685	BTU (mean)
Candle/cm^2	3.146	lamberts
Candle/in^2	0.4870	lamberts
Centares	1.0	m^2
Centigrade (degrees)	(°C × 9/5) + 32	°F
Centigrade (degrees)	°C + 273.18	°K
Centigrams	0.01	grams
Centiliters	0.3382	ounce (fluid) U.S.
Centiliters	0.6103	cubic inch
Centiliters	2.705	drams
Centiliters	0.010	liters
Centimeters	0.03281	feet
Centimeters	0.3937	inches
Centimeters	1×10^{-5}	kilometers
Centimeters	0.01	meters
Centimeters	6.214×10^{-6}	miles
Centimeters	10.	millimeters
Centimeters	393.7	mils
Centimeters	0.001094	yards
Centimeters	1×10^{-4}	microns
Centimeters	1×10^{8}	angstrom units
Centimeter-dynes	0.001020	cn-grams
Centimeter-dynes	1.020×10^{-8}	meter-kgs
Centimeter-dynes	7.376×10^{-8}	lb-ft
Centimeter-grams	980.7	cm-dynes
Centimeter-grams	1×10^{-5}	meter-kg
Centimeter-grams	7.233×10^{-5}	pound-ft
Centimeters of mercury	0.01316	atmospheres
Centimeters of mercury	0.4461	ft of water
Centimeters of mercury	136	kgs/m^2
Centimeters of mercury	27.85	lb/ft^2
Centimeters of mercury	0.1934	lb/in^2
Centimeters/s	1.969	ft/min
Centimeters/s	0.03281	ft/s

Continued

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd

To Convert	Multiply By	To Obtain
Centimeters/s	0.036	km/h
Centimeters/s	0.01934	knots
Centimeters/s	0.60	meters/min
Centimeters/s	0.02237	miles/h
Centimeters/s/s	0.03281	ft/s/s
Centimeters/s/s	0.036	km/h/s
Centimeters/s/s	0.010	meters/s/s
Centimeters/s/s	0.02237	miles/h/s
Centipoise	0.010	gr/cm/s
Centipoise	6.27×10^{-4}	lb/ft-s
Centipoise	2.4	lb/ft-h
Circumference	6.283	radians
Cords	8.0	cord ft
Cord ft	16	cubic ft
Coulombs	2.998×10^{9}	statcoulombs
Coulombs	1.036×10^{-5}	faradays
Coulombs/cm^2	6.452	coulombs/in^2
Coulombs/cm^2	1.0×10^{4}	coulombs/m^2
Coulombs/in^2	0.1550	coulombs/cm^2
Coulombs/in^2	1550	coulombs/m^2
Coulombs/m^2	1.0×10^{-4}	coulombs/cm^2
Coulombs/m^2	6.452×10^{-4}	coulombs/in^2
Cubic centimeters	3.531×10^{-5}	cubic ft
Cubic centimeters	0.06102	cubic in
Cubic centimeters	1.0×10^{-6}	cubic meters
Cubic centimeters	1.308×10^{-6}	cubic yards
Cubic centimeters	2.642×10^{-4}	gallons (U.S. liquid)
Cubic centimeters	0.0010	liters
Cubic centimeters	0.002113	pints (U.S. liquid)
Cubic centimeters	0.001057	quarts (U.S. liquid)
Cubic feet	0.8036	bushels (dry)
Cubic feet	2.8320×10^{4}	cubic cm
Cubic feet	1728	cubic inches
Cubic feet	0.02832	cubic meters
Cubic feet	0.03704	cubic yards
Cubic feet	7.48052	gallons (U.S. liquid)
Cubic feet	28.32	liters
Cubic feet	59.84	pints (U.S. liquid)

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd		
To Convert	**Multiply By**	**To Obtain**
Cubic feet	29.92	quarts (U.S. liquid)
Cubic feet/min	472	cubic cm/s
Cubic feet/min	1.699	cubic m/h
Cubic feet/min	0.1247	gal/s
Cubic feet/min	0.4720	liters/s
Cubic feet/min	62.43	lb water/min
Cubic feet/lb	0.06243	m^3/kg
Cubic feet/lb	62.43	cm^2/g
Cubic feet/s	0.646317	million gal/day
Cubic feet/s	448.831	gal/min
Cubic inches	16.39	cubic cm
Cubic inches	5.787×10^{-4}	cubic ft
Cubic inches	1.639×10^{-5}	cubic meters
Cubic inches	5.787×10^{-4}	cubic ft
Cubic inches	1.639×10^{-5}	cubic meters
Cubic inches	2.143×10^{-5}	cubic yards
Cubic inches	0.004329	gallons
Cubic inches	0.01639	liters
Cubic inches	0.03463	pints (U.S. liquid)
Cubic inches	0.01732	quarts (U.S. liquid)
Cubic meters	28.38	bushels (dry)
Cubic meters	1.0×10^{6}	cubic cm
Cubic meters	35.31	cubic ft
Cubic meters	6.1023×10^{4}	cubic inches
Cubic meters	1.308	cubic yards
Cubic meters	264.2	gallons (U.S. liquid)
Cubic meters	1000	liters
Cubic meters	2113	pints (U.S. liquid)
Cubic meters	1057	quarts (U.S. liquid)
Cubic meters/h	0.5886	cu ft/min
Cubic meters/h at 0°C	0.6223	std cu ft/min at 60°F
Cubic meters/h at 15°C	0.5896	std cu ft/min at 60°F
Cubic meters/kilogram	16.02	cu ft/pound
Cubic yards	7.646×10^{5}	cubic cm
Cubic yards	27	cubic ft
Cubic yards	4.6656×10^{4}	cubic inches
Cubic yards	0.7646	cu meters
Cubic yards	202	gallons (U.S. liquid)

Continued

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd

To Convert	Multiply By	To Obtain
Cubic yards	764.6	liters
Cubic yards	1615.9	pints (U.S. liquid)
Cubic yards	807.9	quarts (U.S. liquid)
Cubic yards/min	0.45	cu ft/s
Cubic yards/min	3.367	gal/s
Cubic yards/min	12.74	1/s
D		
Daltons	1.650×10^{-24}	grams
Days	8.64×10^{4}	seconds
Days	1440	minutes
Days	24.	hours
Decigrams	0.10	grams
Deciliters	0.10	liters
Decimeters	0.10	meters
Degrees (angle)	0.01111	quadrants
Degrees (angle)	0.01745	radians
Degrees (angle)	3600	seconds
Drams (apoth or troy)	0.13714	ounces (avdp.)
Drams (apoth or troy)	0.125	ounces (troy)
Drams (U.S. fluid or apoth)	3.6967	cubic cm
Drams	1.7718	grams
Drams	27.344	grains
Drams	0.0625	ounces
Dynes/cm^2	9.869×10^{-7}	atmospheres
Dynes/cm^2	2.953×10^{-5}	in. of mercury (at 0°C)
Dynes/cm^2	4.015×10^{-4}	in. of water (at 4°C)
Dynes	0.001020	grams
Dynes	1.0×10^{-7}	joules/cm
Dynes	1.0×10^{-5}	joules/cm(newtons)
Dynes	1.020×10^{-6}	kilograms
Dynes	7.233×10^{-5}	poundals
Dynes	2.248×10^{-6}	pounds
Dynes/cm^2	1.0×10^{-6}	bars
E		
Ell	114.30	cm
Ell	45	inches
Em, pica	0.167	inch
Em, pica	0.4233	cm

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd		
To Convert	**Multiply By**	**To Obtain**
Erg/s	1.0	dyne-cm/s
Ergs	9.486×10^{-11}	BTU
Ergs	1.0	dyne-centimeters
Ergs	7.376×10^{-8}	foot-pounds
Ergs	2.389×10^{-8}	gram-calories
Ergs	0.001020	gram-centimeters
Ergs	3.7250×10^{-14}	horsepower-h
Ergs	1.0×10^{-7}	joules
Ergs	2.389×10^{-11}	kg-calories
Ergs	1.020×10^{-8}	kg-meters
Ergs	2.773×10^{-14}	kilowatt-h
Ergs	2.773×10^{-11}	watt-h
Ergs/s	5.668×10^{-9}	BTU/min
Ergs/s	4.426×10^{-6}	ft lb/min
Ergs/s	7.3756×10^{-8}	ft lb/s
Ergs/s	1.341×10^{-10}	horsepower
Ergs/s	1.433×10^{-9}	kg-calories/min
Ergs/s	1×10^{-10}	kilowatts
F		
Farads	1×10^{6}	microfarads
Faraday/s	9.65×10^{4}	ampere (absolute)
Faradays	26.8	ampere-hours
Faradays	9.649×10^{4}	coulombs
Fathoms	1.8228	meters
Fathoms	6.0	feet
Feet	30.48	centimeters
Feet	3.048×10^{-4}	kilometers
Feet	0.3048	meters
Feet	1.645×10^{-4}	miles (naut.)
Feet	1.894×10^{-4}	miles (stat.)
Feet	304.8	millimeters
Feet	1.2×10^{4}	mils
Feet of water	0.0295	atmospheres
Feet of water	0.8826	in. of mercury
Feet of water	0.03048	kg/cm^2
Feet of water	0.4335	lb/in^2
Feet/s	30.48	cm/s
Feet/s	1.097	km/h

Continued

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd

To Convert	Multiply By	To Obtain
Feet/s	0.5921	knots
Feet/s	0.6818	miles/h
Feet/s/s	30.48	cm/s/s
Feet/s/s	1.097	km/h/s
Feet/s/s	0.3048	meters/s/s
Feet/s/s	0.6818	miles/h/s
Foot-pounds	0.00128508	BTU
Foot-pounds	0.3241	gram-calories
Foot-pounds	5.0505×10^{-7}	horsepower-h
Foot-pounds	1.35582	joules
Foot-pounds	3.241×10^{-4}	kg-calories
Foot-pounds	0.1383	kg-meters
Foot-pounds	3.766×10^{-7}	kilowatt-h
Foot-pounds/min	0.001286	BTU/min
Foot-pounds/min	0.01667	ft-lb/s
Foot-pounds/min	3.030×10^{-5}	horsepower
Foot-pounds/min	3.241×10^{-4}	kg-calories/min
Foot-pounds/min	2.260×10^{-5}	kilowatts
Foot-pound$_f$/pound$_m$	2.989	newton-meters/kg
Foot-pound$_f$/pound$_m$	0.3048	meter-kg$_f$/kg$_m$
Foot-pound$_f$/pound$_m$	0.002989	kilonewton-m/kg
Foot-pounds/s	4.6263	BTU/h
Foot-pounds/s	0.07717	BTU/min
Foot-pounds/s	0.001818	horsepower
Foot-pounds/s	0.01945	kg-calories/min
Foot-pounds/s	0.001356	kilowatts
Furlongs	0.125	miles (U.S.)
Furlongs	40	rods
Furlongs	660	feet
Furlongs	201.17	meters
G		
Gallons	3785	cubic cm
Gallons	0.1337	cubic feet
Gallons	231	cubic inches
Gallons	0.003785	cubic meters
Gallons	0.004951	cubic yards
Gallons	3.785	liters
Gallons (liq. Br. Imp.)	1.20095	gallons (U.S. liquid)

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd		
To Convert	**Multiply By**	**To Obtain**
Gallons (U.S.)	0.83267	gallons (imp.)
Gallons of water	8.337	pounds of water
Gallons/min	0.002228	ft^3/s
Gallons/min	0.06308	liters/s
Gallons/min	8.0208	ft^3/h
Grams	980.7	dynes
Grams	15.43	grains (troy)
Grams	9.807×10^{-5}	joules/cm
Grams	0.009807	joules/meter (newtons)
Grams	0.0010	kilograms
Grams	1000	milligrams
Grams	0.03527	ounces (avdp.)
Grams	0.03515	ounces (troy)
Grams	0.07093	poundals
Grams	0.002205	pounds
Grams/cm	0.0056	lb/in
$Grams/cm^3$	0.03613	lb/in^3
Grams/liter	0.062427	lb/ft^3
Gram-calories	0.0039683	BTU
Gram-calories	4.184×10^{7}	ergs
Gram-calories	3.086	foot-pounds
Gram-calories	1.5596×10^{-6}	horsepower-h
Gram-calories	1.162×10^{-6}	kilowatt-h
Gram-calories	0.001162	watt-h
Gram-calories/s	14.286	BTU/h
Gram-centimeters	9.297×10^{-8}	BTU
Gram-centimeters	980.7	ergs
Gram-centimeters	9.807×10^{-5}	joules
Gram-centimeters	2.343×10^{-8}	kg-calories
H		
Hand	10.16	cm
Hectares	2.471	acres
Hectares	1.076×10^{5}	square feet
Hectograms	100	grams
Hectoliters	100	liters
Hectometers	100	meters
Hectowatts	100	watts
Henries	1000	millihenries

Continued

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd

To Convert	Multiply By	To Obtain
Hogsheads (British)	10.114	cubic ft
Hogsheads (U.S.)	8.42184	cubic ft
Hogsheads (U.S.)	63	gallons (U.S.)
Horsepower	42.44	BTU/min
Horsepower	3.3×10^4	foot-lb/min
Horsepower	550	foot-lb/s
Horsepower (metric)	0.9863	horsepower
Horsepower	1.014	horsepower (metric)
Horsepower	10.68	kg-calories/min
Horsepower	0.7457	kilowatts
Horsepower	745.7	watts
Horsepower (boiler)	9.803	kilowatts
Horsepower-hours	2544.47	BTU
Horsepower-hours	2.6845×10^{13}	egrs
Horsepower-hours	1.98×10^6	foot-lbs
Horsepower-hours	6.4119×10^5	gram-calories
Horsepower-hours	2.684×10^6	joules
Horsepower-hours	641.7	kg-calories
Horsepower-hours	2.737×10^5	kg-meters
Horsepower-hours	0.7457	kilowatt-h
Hours	0.04167	days
Hours	0.005952	weeks
Hours	3600	seconds
Hundredwgts (long)	112	pounds
Hundredwgts (long)	0.050	tons (long)
Hundredwgts (long)	50.8023	kilograms
Hundredwgts (short)	0.045359	tons (metric)
Hundredwgts (short)	0.0446429	tons (long)
Hundredwgts (short)	45.3592	kilograms
I		
Inches	2.540	centimeters
Inches	0.02540	meters
Inches	1.578×10^{-5}	miles
Inches	25.4	millimeters
Inches	1000	mils
Inches	0.02778	yards
Inches	2.54×10^8	angstrom units
Inches	0.0050505	rods

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd

To Convert	Multiply By	To Obtain
Inches of mercury	0.03342	atmospheres
Inches of mercury	1.1340	feet of water
Inches of mercury	0.0345316	kg/cm^2
Inches of mercury	345.316	kg/m^2
Inches of mercury	70.73	lb/ft^2
Inches of mercury	0.491154	lb/in^2
Inches of mercury	0.0338639	bar
In. of water (at 4°C)	0.002458	atmospheres
In. of water (at 4°C)	0.07355	inches of mercury
In. of water (at 4°C)	0.00254	kg/cm^2
In. of water (at 4°C)	0.5781	$ounces/in^2$
In. of water (at 4°C)	5.204	lb/ft^2
In. of water (at 4°C)	0.03613	lb/in^2
In. of water (at 4°C)	0.2489	kPa
J		
Joules	9.468×10^{-4}	BTU
Joules	1.0×10^{7}	ergs
Joules	0.7736	foot-pounds
Joules	2.389×10^{-4}	kg-calories
Joules	0.1020	kg-meters
Joules	2.778×10^{-4}	watt-h
Joules/cm	1.020×10^{4}	grams
Joules/cm	1.0×10^{7}	dynes
Joules/cm	100	joules/meter (newtons)
Joules/cm	723.3	poundals
Joules/cm	22.48	pounds
K		
Kilograms	9.80665×10^{5}	dynes
Kilograms	1000	grams
Kilograms	0.09807	joules/cm
Kilograms	9.807	joules/meter (newtons)
Kilograms	70.93	poundals
Kilograms	2.2046	pounds
Kilograms	9.842×10^{-4}	tons (long)
Kilograms	0.001102	tons (short)
Kilograms	35.274	ounces (avdp.)
kg/m^3	0.0010	$grams/cm^3$
kg/m^3	0.06243	lb/ft^3

Continued

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd

To Convert	Multiply By	To Obtain
kg/m^3	3.613×10^{-5}	lb/in^3
kg/m	0.672	lb/ft
kg/cm^2	9.80665×10^5	$dynes/cm^2$
kg/cm^2	0.967841	atmospheres
kg/cm^2	32.841	feet of water
kg/cm^2	28.9590	inches of mercury at 32°F
kg/cm^2	2048	lb/ft^2
kg/cm^2	14.2233	lb/in^2
kg/cm^2	0.9807	bars
kg/m^2	9.678×10^{-5}	atmospheres
kg/m^2	9.807×10^{-5}	bars
kg/m^2	0.003281	feet of water
kg/m^2	0.002896	inches of mercury
kg/m^2	0.2048	lb/ft^2
kg/m^2	0.001422	lb/in^2
kg/m^2	98.0665	$dynes/cm^2$
kg/mm^2	1.0×10^6	kg/m^2
Kilogram-calories	3.968	BTU
Kilogram-calories	3086	foot-pounds
Kilogram-calories	0.001558	horsepower-h
Kilogram-calories	4183	joules
Kilogram-calories	426.9	kg-meters
Kilogram-calories	4.186	kilojoules
Kilogram-calories	0.001163	kilowatt-h
Kilogram-calories/min	51.43	ft-lbs/s
Kilogram-calories/min	0.09351	horsepower
Kilogram-calories/min	0.06972	kilowatts
Kilograms/hour	0.03674	lb/min
Kilograms/hour	2.205	lb/h
Kilogram-meters	0.009295	BTU
Kilogram-meters	9.807×10^7	ergs
Kilogram-meters	7.23301	foot-pounds
Kilogram-meters	9.80665	joules
Kilogram-meters	0.002342	kg-calories
Kilogram-meters	2.723×10^{-6}	kilowatt-h
Kilojoule/kilogram	0.4299	BTU/lb_m
Kilolines	1000	maxwells
Kiloliters	1000	liters

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd

To Convert	Multiply By	To Obtain
Kiloliters	1.308	cubic yards
Kiloliters	26.316	cubic feet
Kiloliters	264.18	gallons (U.S. liquids)
Kilometers	1.0×10^5	centimeters
Kilometers	3281	feet
Kilometers	3.937×10^4	inches
Kilometers	1000	meters
Kilometers	0.6214	miles (statute)
Kilometers	0.5396	miles (nautical)
Kilometers	1.0×10^6	millimeters
Kilometers	1093.6	yards
Kilometers/h	27.78	cm/s
Kilometers/h	54.68	ft/min
Kilometers/h	0.9113	ft/s
Kilometers/h	0.5396	knots
Kilometers/h	16.67	meters/min
Kilometers/h	0.6214	miles/h
Kilometers/h/s	27.78	cm/s/s
Kilometers/h/s	0.9113	ft/s/s
Kilometers/h/s	0.2778	meters/s/s
Kilometers/h/s	0.6214	miles/h/s
Kilonewton-meters/kilogram	334.6	ft-lb_f/lb_m
Kilopascals	0.145	lb/in^2
Kilowatts	56.92	BTU/min
Kilowatts	4.426×10^4	foot-lb/min
Kilowatts	737.6	foot-lb/s
Kilowatts	1.341	horsepower
Kilowatts	14.34	kg-calories/min
Kilowatts	1000	watts
Kilowatt-h	3413	BTU
Kilowatt-h	3.6×10^{13}	ergs
Kilowatt-h	2.655×10^6	foot-lb
Kilowatt-h	8598.5	gram calories
Kilowatt-h	1.341	horsepower-hours
Kilowatt-h	3.6×10^6	joules
Kilowatt-h	860.5	kg-calories
Kilowatt-h	3.671×10^5	kg-meters
Kilowatt-h	3.53	pounds of water evaporated from and at 212°F

Continued

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd

To Convert	Multiply By	To Obtain
Kilowatt-h	22.75	pounds of water raised from 62° to 212°F
Knots	6080	feet/h
Knots	1.8532	kilometers/h
Knots	1.0	nautical miles/h
Knots	1.151	statute miles/h
Knots	2027	yards/h
Knots	1.689	feet/s
Knots	51.48	cm/s
L		
Lambert	0.3183	candle/cm^2
Lambert	2.054	candle/in^2
League	3.0	miles (approx.)
Light year	5.9×10^{12}	miles
Light year	9.46091×10^{12}	kilometers
Liters	0.02838	bushels (U.S. dry)
Liters	1000	cubic cm
Liters	0.03531	cubic ft
Liters	61.02	cubic inches
Liters	0.0010	cubic meters
Liters	0.001308	cubic yards
Liters	0.2642	gallons (U.S. liquid)
Liters	2.113	pints (U.S. liquid)
Liters	1.057	quarts (U.S. liquid)
Liters/min	5.886×10^{-4}	ft^3/s
Liters/min	0.004403	gal/s
Log10*n*	2.303	In n
In *n*	0.4343	log10n
Lumen	0.07958	spherical candle power
Lumen/ft^2	1.0	foot-candles
Lumen/ft^2	10.76	lumen-m^2
Lux	0.0929	foot-candles
M		
Maxwells	0.0010	kilolines
Maxwells	1.0×10^{-8}	webers
Mega pascal, MPa	145	pounds/in^2
Mega watt	1341.0	horsepower
Meters	1.0×10^{10}	angstrom units
Meters	100	centimeters

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd

To Convert	Multiply By	To Obtain
Meters	0.54681	fathoms
Meters	3.281	feet
Meters	39.37	inches
Meters	0.0010	kilometers
Meters	5.396×10^{-4}	miles (nautical)
Meters	6.214×10^{-4}	miles (statute)
Meters	1000	millimeters
Meters	1.094	yards
Meters/min	1.667	cm/s
Meters/min	3.281	feet/min
Meters/min	0.05468	feet/s
Meters/min	0.060	km/h
Meters/min	0.03238	knots
Meters/min	0.03728	miles/h
Meters/s	196.8	feet/min
Meters/s	3.281	feet/s
Meters/s	3.6	kilometers/h
Meters/s	0.060	kilometers/min
Meters/s	2.237	miles/h
Meters/s	0.03728	miles/min
Meters/s/s	100	cm/s/s
Meters/s/s	3.281	ft/s/s
Meters/s/s	3.6	km/h/s
Meters/s/s	2.237	miles/h/s
Meter-kilograms	9.807×10^{7}	cm-dynes
Meter-kilograms	1.0×10^{5}	cm/gram
Meter-kilograms	7.233	foot-pound
Meter-kilograms$_f$/kilogram$_m$	3.281	foot-pounds$_F$/pound$_m$
Micromicrons	1.0×10^{-12}	meters
Microns	1.0×10^{-6}	meters
Microns	25.4	mils
Miles (nautical)	6076	feet
Miles (nautical)	1.853	kilometers
Miles (nautical)	1853	meters
Miles (nautical)	1.1516	miles (statute)
Miles (nautical)	2025.4	yards
Miles (statute)	1.609×10^{5}	centimeters
Miles (statute)	5280	feet

Continued

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd

To Convert	Multiply By	To Obtain
Miles (statute)	6.336×10^{4}	inches
Miles (statute)	1.609	kilometers
Miles (statute)	1609	meters
Miles (statute)	0.8684	miles (nautical)
Miles (statute)	1760	yards
Miles (statute)	1.69×10^{-13}	light years
Miles/h	44.70	cm/s
Miles/h	88	ft/min
Miles/h	1.467	ft/s
Miles/h	1.6093	km/h
Miles/h	0.02682	km/min
Miles/h	0.8684	knots
Miles/h	26.82	meters/min
Milligrams	0.0010	grams
Milliliters	0.0010	liters
Millimeters	0.10	centimeters
Millimeters	0.003281	feet
Millimeters	0.03937	inches
Millimeters	1.0×10^{-6}	kilometers
Millimeters	0.0010	meters
Millimeters	6.214×10^{-7}	miles
Millimeters	39.37	mils
Millimeters	0.001094	yards
Million gal/day	1.54723	ft^3/s
Mils	0.00254	centimeters
Mils	8.333×10^{-5}	feet
Mils	0.0010	inches
Mils	2.54×10^{-8}	kilometers
Mils	2.778×10^{-5}	yards
Mils	0.03937	microns
N		
Newtons	1.0×10^{5}	dynes
Newtons	0.2248	pounds
Newton-meter/kilogram	0.3346	$ft\text{-}lb_f/lb_m$
Newtons/m^3	1.45×10^{-4}	lb/in^2
Newtons/m^3	1.0×10^{-5}	bars
Newton-meters	0.7366	

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd		
To Convert	**Multiply By**	**To Obtain**
O		
Ohms	1.0×1.0^{6}	microhms
Ounces	8.0	drams
Ounces	437.5	grains
Ounces	28.349	grams
Ounces	0.0625	pounds
Ounces	0.9115	ounces/(troy)
Ounces	2.790×10^{-5}	tons (long)
Ounces	3.125×10^{-5}	tons (short)
Ounces (fluid)	1.805	cubic inches
Ounces (fluid)	0.02957	liters
Ounces (troy)	480	grains
Ounces (troy)	31.103	grams
Ounces (troy)	1.097	ounces (avdp.)
Ounces (troy)	20	pennyweights (troy)
Ounces (troy)	0.08333	pounds (troy)
Ounce/in^2	4309	dynes/cm^2
Ounce/in^2	0.0625	pounds/in^2
P		
Parts/million	0.0584	grains/U.S. gal
Parts/million	0.07016	grains/imp. gal
Parts/million	8.345	pounds/million gal
Pascals, N/m^3	1.45×10^{-4}	lb/in^2
Pascals, N/m^3	10^{-5}	bars
Pecks (British)	554.6	cubic inches
Pecks (British)	9.0919	liters
Pecks (U.S.)	0.25	bushels
Pecks (U.S.)	537.6	cubic inches
Pecks (U.S.)	8.8096	liters
Pecks (U.S.)	8	quarts/dry
Pennyweights (troy)	24	grains
Pennyweights (troy)	0.050	ounces (troy)
Pennyweights (troy)	1.555	grams
Pennyweights (troy)	0.0041667	pounds (troy)
Pints (dry)	33.6	cubic inches
Pints (dry)	0.015625	cubic inches
Pints (dry)	0.50	quarts

Continued

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd		
To Convert	**Multiply By**	**To Obtain**
Pints (dry)	0.55059	liters
Pints (liquid)	473.2	cubic cm
Pints (liquid)	0.01671	cubic ft
Pints (liquid)	28.87	cubic inches
Pints (liquid)	4.732×10^{-4}	cubic meters
Pints (liquid)	6.189×10^{-4}	cubic yards
Pints (liquid)	0.125	gallons
Pints (liquid)	0.4732	liters
Pints (liquid)	0.50	quarts (liquid)
Planck's quantum	6.624×10^{-27}	erg-seconds
Poise	1.0	gram/cm-s
Pounds (avdp.)	14.583	ounces (troy)
Poundals	1.3826×10^{4}	dynes
Poundals	14.1	grams
Poundals	0.001383	joules/cm
Poundals	0.1383	joules/meter (newtons)
Poundals	0.0141	kilograms
Poundals	0.03108	pounds
Pounds	256	drams
Pounds	4.448×10^{5}	dynes
Pounds	7000	grains
Pounds	453.59	grams
Pounds	0.04448	joules/cm
Pounds	4.448	joules/meter (newtons)
Pounds	0.4536	kilograms
Pounds	16	ounces
Pounds	14.58	ounces (troy)
Pounds	32.17	poundals
Pounds	1.21528	pounds (troy)
Pounds	5.0×10^{-4}	tons (short)
Pounds (troy)	5760	grains
Pounds (troy)	373.24	grams
Pounds (troy)	13.166	ounces (avdp.)
Pounds (troy)	12	ounces (troy)
Pounds (troy)	240	pennyweights (troy)
Pounds (troy)	0.82286	pounds (avdp.)
Pounds (troy)	3.6735×10^{-4}	tons (long)
Pounds (troy)	3.7324×10^{-4}	tons (metric)
Pounds (troy)	4.1143×10^{-4}	tons (short)

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd

To Convert	Multiply By	To Obtain
Pounds of water	0.01602	cubic ft
Pounds of water	27.68	cubic inches
Pounds of water	0.1198	gallons
Pounds of water/min	2.670×10^{-4}	ft^3/s
Pound-feet	1.356×10^{7}	cm-dynes
Pound-feet	1.3825×10^{4}	cm-grams
Pound-feet	0.1383	meter-kg
Pounds/ft^3	0.01602	grams/cm^3
Pounds/ft^3	16.02	kg/m^3
Pounds/ft^3	5.787×10^{-4}	lb/in^3
Pounds/ft^3	5.456×10^{-9}	lb/mil-foot
Pounds/in^3	27.68	grams/cm^3
Pounds/in^3	2.768×10^{4}	kg/m^3
Pounds/in^3	1728	lb/ft^3
Pounds/in^3	9.425×10^{-6}	lb/mil-foot
Pounds/ft	1.488	kg/meter
Pounds/in	178.6	grams/cm
Pounds/min	27.22	kilograms/h
Pounds/h	0.4536	kilograms/h
Pounds/mil-foot	2.306×10^{6}	grams/cm^3
Pounds/ft^2	4.725×10^{-4}	atmospheres
Pounds/ft^2	0.01602	feet of water
Pounds/ft^2	0.01414	inches of mercury
Pounds/ft^2	4.882	kg/$meter^2$
Pounds/ft^2	0.006944	pounds/$inch^2$
Pounds/in^2	0.0680460	atmospheres
Pounds/in^2	0.0689476	bar
Pounds/in^2	2.3089	feet of water
Pounds/in^2	2.03602	inches of mercury at 32°F
Pounds/in^2	703.1	kg/m^2
Pounds/in^2	144	lb/ft^2
Pounds/in^2	0.0703070	kg/cm^2
Pounds/in^2	0.006895	mega pascal (MPa)
Pounds/in^2	6.895	kilo pascal (kPa), kN/m^2
Pounds/in^2	27.707	in. of water (at 4°C)
Q		
Quadrants (angle)	90	degrees
Quadrants (angle)	5400	minutes

Continued

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd

To Convert	Multiply By	To Obtain
Guadrants (angle)	1.571	radians
Quadrants (angle)	3.24×10^5	seconds
Quarts (dry)	67.2	cubic inches
Quarts (liquid)	946.4	cubic cm
Quarts (liquid)	0.03342	cubic ft
Quarts (liquid)	57.75	cubic inches
Quarts (liquid)	0.464×10^{-4}	cubic meters
Quarts (liquid)	0.001238	cubic yards
Quarts (liquid)	0.25	gallons
Quarts (liquid)	0.9463	liters
R		
Radians	57.296	degrees
Radians	3438	minutes
Radians	0.6366	quadrants
Radians	2.063×10^5	seconds
Radians/s	57.296	degrees/s
Radians/s	9.549	revolutions/min
Radians/s	0.1592	revolutions/s
Radians/s/s	572.96	revs/min/min
Radians/s/s	9.549	revs/min/s
Radians/s/s	0.1592	revs/s/s
Reams	500	sheets
Revolutions	360	degrees
Revolutions	4.0	quadrants
Revolutions	6.283	radians
Revolutions/min	6.0	degrees/s
Revolutions/min	0.1047	radians/s
Revolutions/min	0.01667	revs/s
Rods	0.25	chains (gunters)
Rods	5.029	meters
Rods (surveyors' meas.)	5.5	yards
Rods	16.5	feet
Rods	198	inches
Rods	0.003125	miles
Rope	20	feet
S		
Slugs	14.59	kilograms
Slugs	32.17	pounds

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd

To Convert	Multiply By	To Obtain
Square centimeters	0.001076	square feet
Square centimeters	0.1550	square inches
Square centimeters	1.0×10^{-4}	square meters
Square centimeters	3.861×10^{-11}	square miles
Square centimeters	100	square millimeters
Square centimeters	1.196×10^{-4}	square yards
Square degrees	3.0462×10^{-4}	steradians
Square feet	2.296×10^{-5}	acres
Square feet	1.883×10^{8}	circular mils
Square feet	929	square cm
Square feet	144	square inches
Square feet	0.0929	square meters
Square feet	3.587×10^{-8}	square miles
Square feet	9.29×10^{4}	square millimeters
Square feet	0.1111	square yards
Square inches	1.273×10^{6}	circular mils
Square inches	6.452	square cm
Square inches	0.006944	square ft
Square inches	645.2	square millimeters
Square inches	1.0×10^{6}	square mils
Square inches	7.716×10^{-4}	square yards
Square kilometers	247.1	acres
Square kilometers	1.0×10^{10}	square cm
Square kilometers	1.076×10^{7}	square ft
Square kilometers	1.550×10^{9}	square inches
Square kilometers	1.0×10^{6}	square meters
Square kilometers	0.3861	square miles
Square kilometers	1.196×10^{6}	square yards
Square meters	2.471×10^{-4}	acres
Square meters	1.0×10^{4}	square cm
Square meters	10.76	square ft
Square meters	1550	square inches
Square meters	3.861×10^{-7}	square miles
Square meters	1.0×10^{6}	square millimeters
Square meters	1.196	square yards
Square miles	640	acres
Square miles	2.788×10^{7}	square ft

Continued

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd		
To Convert	**Multiply By**	**To Obtain**
Square miles	2.590	square km
Square miles	2.590×10^{6}	square meters
Square miles	3.098×10^{6}	square yards
Standard ft^3/min at 60°F	1.607	m^3/h at 0°C
Standard ft^3/min at 60°F	1.696	m^3/h at 15°C
Square millimeters	1973	circular mils
Square millimeters	0.010	square cm
Square millimeters	1.076×10^{-5}	square ft
Square millimeters	0.00155	square inches
Square mils	1.273	circular mils
Square mils	6.452×10^{-6}	square cm
Square mils	1.0×10^{-6}	square inches
Square yards	2.066×10^{-4}	acres
Square yards	8361	square cm
Square yards	9.0	square ft
Square yards	1296	square inches
Square yards	0.8361	square meters
Square yards	3.228×10^{-7}	square miles
Square yards	8.361×10^{5}	square millimeters
T		
Temperature (°C) + 273	1.0	absolute temperature (°K)
Temperature (°C) + 17.78	1.8	temperature (°F)
Temperature (°F) + 460	1.0	absolute temperature (°R)
Temperature (°F) − 32	5/9	temperature (°C)
Tonne	1000	kilograms
Tonne	2205	pounds
Tonne	1.0	(metric) tons
Tons (long)	1016	kilograms
Tons (long)	2240	pounds
Tons (long)	1.12	tons (short)
Tons (metric)	1000	kilograms
Tons (metric)	2205	pounds
Tons (short)	907.18	kilograms
Tons (short)	3.2×10^{4}	ounces
Tons (short)	2.9166×10^{4}	ounces (troy)
Tons (short)	2000	pounds
Tons (short)	2430	pounds (troy)
Tons (short)	0.89287	tons (long)

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd		
To Convert	**Multiply By**	**To Obtain**
Tons (short)	0.9078	tons (metric)
Tons (short)/ft^2	9765	kg/m^2
Tons (short)/ft^2	13.89	lb/in^2
Tons (short)/in^2	1.406×10^6	kg/m^2
Tons (short)/in^2	2000	lb/in^2
W		
Watts	3.4129	BTU/h
Watts	0.05688	BTU/min
Watts	1.0×10^7	ergs/s
Watts	44.27	ft-lb/min
Watts	0.7378	ft-lb/s
Watts	0.001341	horsepower
Watts	0.00136	horsepower (metric)
Watts	0.01433	kg-calories/min
Watts	0.0010	kilowatts
Watts (abs.)	1.0	joules/s
Watt-hours	3.413	BTU
Watt-hours	3.6×10^{10}	ergs
Watt-hours	2656	foot-lb
Watt-hours	860.5	gram-calories
Watt-hours	0.001341	horsepower-hours
Watt-hours	0.8605	kilogram-calories
Watt-hours	367.2	kilogram-meters
Watt-hours	0.0010	kilowatt-hours
Webers	1.0×10^8	maxwells
Webers	1.0×10^5	kilolines
Webers/in^2	1.55×10^7	gausses
Webers/in^2	1.0×10^8	lines/in^2
Webers/in^2	0.155	webers/cm^2
Webers/in^2	1550	webers/m^2
Webers/m^2	1.0×10^4	gausses
Webers/m^2	6.452×1.0^4	lines/in^2
Webers/m^2	$1.0 \times l.0^{-4}$	webers/cm^2
Webers/m^2	6.452×10^{-4}	webers/in^2
Weeks	168	hours
Weeks	1.008×10^4	minutes
Weeks	6.048×10^5	seconds

Continued

A Table of Conversion Factors is Provided in Alphabetical Order—cont'd		
To Convert	**Multiply By**	**To Obtain**
Y		
Yards	91.44	centimeters
Yards	9.144×10^{-4}	kilometers
Yards	0.9144	meters
Yards	4.934×10^{-4}	miles (nautical)
Yards	5.682×10^{-4}	miles (statute)
Yards	914.4	millimeters
Years	365.256	days (mean solar)
Years	8766.1	hours (mean solar)

Engineering Constants	
1 HP = 33,000 ft-lb/min	1 radian = 57.296 degrees
1 HP = .7457 K.W. = K.W./1.341	1 meter = 100 cm = 39.37 inches
1 HP = 2546.4 BTU per hour	1 kilometer = 0.62137 miles
1 BTU = Heat required to raise 1 lb water 1°F	1 gallon = 231 cubic inches
1 BTU = 778.16 Foot-pounds	1 barrel = 31.5 gallons
1 kilowatt hour = 3413 BTU	Atmospheric pressure = 14.7 pounds per square inch = 29.92 inches mercury at 32°F
Heat value of carbon = 14,600 BTU per pound	1 pound per square inch pressure = 2.3095 feet fresh water at 62°F = 2.0355 inches mercury at 32°F = 2.0416 inches mercury at 62°F
Latent heat of fusion of ice = 143.15 BTU per pound	
Latent heat of evaporation of water at 212°F = 970.4 BTU per pound	Water pressure (pounds per square inch) = 0.433 × height of water in feet (fresh water at 62°F)
Total heat of saturated steam at atmospheric pressure = 1150.4 BTU per pound	Weight of 1 cubic foot fresh water = 62.355 lb at 62°F = 59.76 lb at 212°F
1 ton of refrigeration = 288,000 BTU per 24 h	Weight of 1 cubic foot air at 14.7 lb per square inch pressure = 0.07608 lb at 62°F = 0.08703 lb at 32°F
Gas constant (R)	
$R = 1545.32 \dfrac{\text{ft} - \text{lb}}{\text{Lb} - \text{Mole} - {}^\circ\text{R}}$	Velocity of sound in dry air at 0°C and 1 atm = 33,136 cm/s = 1089 ft/s
$R = 8.314 \dfrac{\text{joules}}{\text{g} - \text{Mole} - {}^\circ\text{K}}$	Acceleration of gravity (standard) = 32.17 ft/s^2 = 980.6 cm/s^2

Appendix D

Permissible Deviations and Fluctuations

Permissible Deviation From Specified Operating Conditions	
Inlet pressure	5%
Inlet temperature	8%
Speed	2%
Molecular weight	2%
Cooling temperature difference	5%
Coolant flow rate	3%
Capacity	4%

Permissible Deviation From Specified Operating Parameters	
Specific volume ratio	95%–105%
Flow coefficient	96%–104%
Mach #	See Figs. D.1 and D.2
Re #	See Fig. D.3

Permissible Fluctuation of Test Readings	
Inlet pressure	2%
Inlet temperature	0.5%
Discharge pressure	2%
Nozzle differential pressure	2%
Nozzle temperature	0.5%
Speed	0.5%
Torque	1%
Electric motor power	1%
MW	0.25%
Cooling water inlet temperature	0.5%
Cooling water flow rate	2%
Line voltage	2%

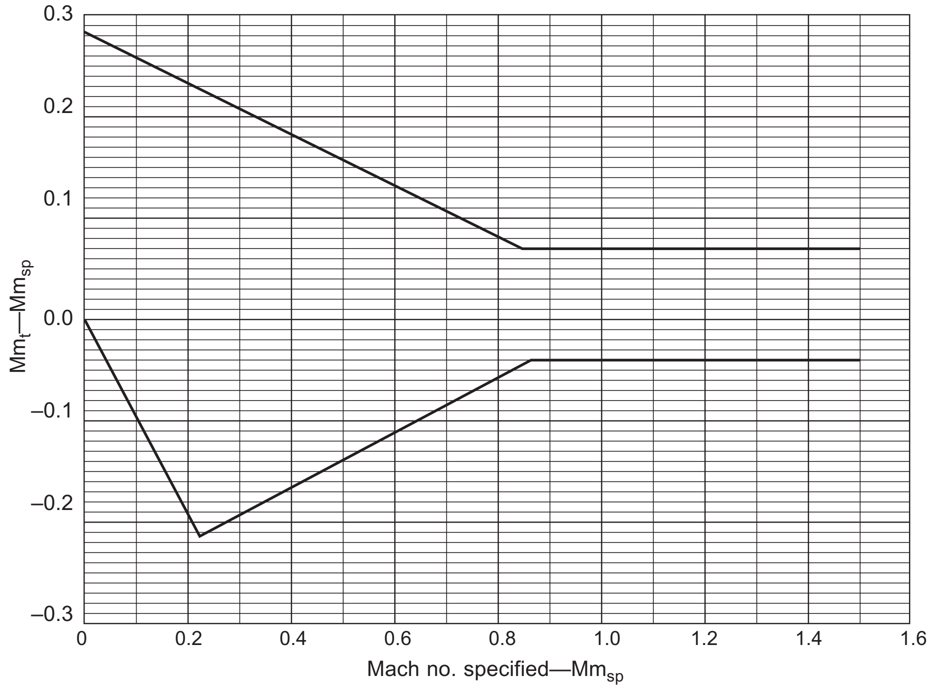


FIG. D.1 Allowable machine Mach number departures, centrifugal compressors.

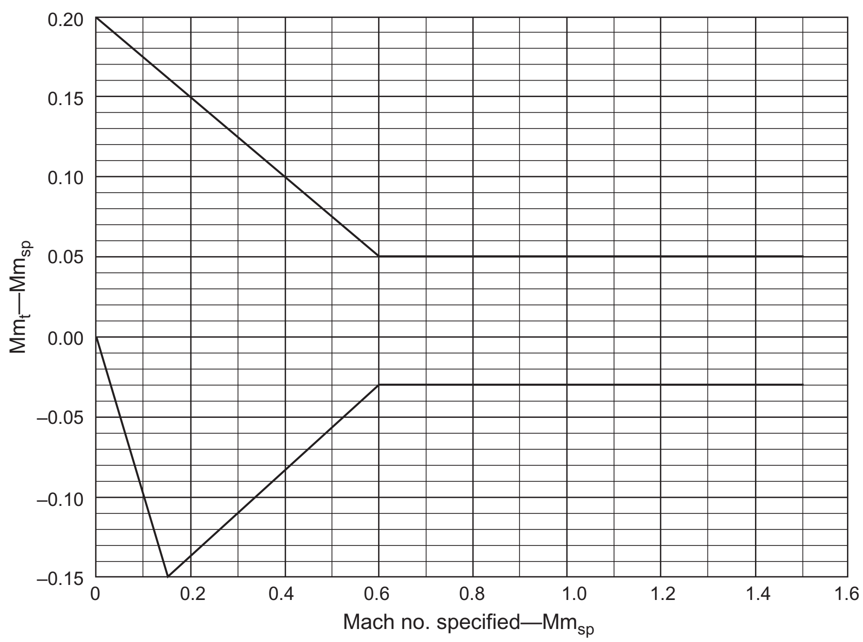


FIG. D.2 Allowable machine Mach number departures, axial compressors.

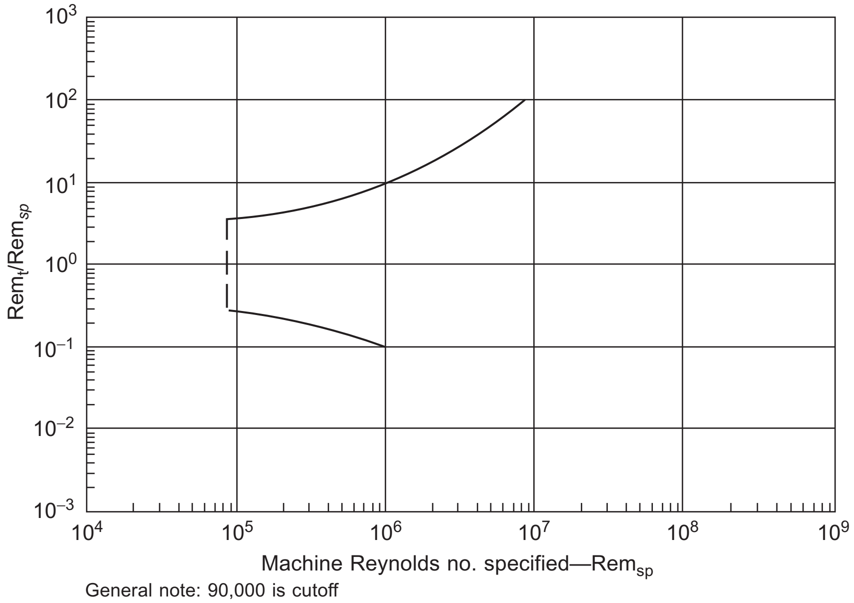


FIG. D.3 Allowable machine Reynolds number departures, centrifugal compressors.

This page intentionally left blank

Appendix E

Thermal Expansion Factor

TABLE E.1 F_{AA}—Thermal Expansion Factor for Annubar® Fluid Flow Meters

Alum	Copper	Type 430	2% CRMO	5% CRMO	Bronze	Steel	Monel	Type 316 or Type 304	Corr Factor, F_{AA}
								0.992	
–264					–317				0.993
–204	–322				–245				0.994
–155	–230				–190			–276	0.995
–108	–163				–137		–236	–189	0.996
–63	–102				–86		–150	–119	0.997
–19	–44				–34		–71	–55	0.998
+25	+19	+44	–13	–14	+17	–6	+2	+7	0.999
+68	+68	+68	+68	+68	+68	+68	+68	+68	1.000
+113	+127	+157	+146	+151	+122	+144	+136	+130	1.001
		+246	+222	+232	+175	+218	+199	+186	1.002
		+332	+296	+312	+225	+289	+260	+240	1.003
		+415	+366	+389	+273	+358	+319	+292	1.004
		+494	+434	+460	+321	+425	+377	+343	1.005
		+568	+501	+527	+369	+489	+433	+391	1.006
		+641	+566	+594	+417	+551	+489	+439	1.007
		+713	+629	+662		+613	+544	+488	1.008
		+783	+690	+730		+675	+599	+536	1.009
		+851	+750	+795		+735	+653	+584	1.010
		+918	+811	+858		+794	+717	+631	1.011
		+986	+871	+918		+851	+759	+674	1.012
		+1054	+928	+979		+907	+810	+727	1.013
		+1121	+984	+1040		+961	+861	+777	1.014
		+1189	+1038	+1102		+1015	+911	+799	1.015

Courtesy: Dieterich Standard, Annubar® Flow Hand book, Boulder, CO, 1997 [34].

This page intentionally left blank

Appendix F

Surge Identification

The following is the preferred procedure for establishing the location of the surge point. Before using this procedure or any other procedure in this book, you should first check with the compressor manufacturer.

(1) Slowly close the recycle or blow-off valve, while monitoring the following parameters:
  a. Blow-off or recycle valve position, % open.
  b. Audible sound level at the inlet of the compressor. Listen for a pulsing sound.
  c. Audible sound level at the compressor discharge. Listen for a pulsing sound, a low frequency 0–25 Hz.
  d. Compressor suction pressure immediately upstream of the compressor inlet flange. Monitor both the local pressure gage (for low pressure, a water manometer works well) and the pressure transmitter. Watch for a bouncing in the pressure level. Note that the pressure transmitter may not show this unless it is rated as a dynamic device, with a rise time below 0.1 s. When the dynamic amplitude exceeds 20% of the gage static pressure or the compressor pressure rise, consider the unit to be in surge.
  e. Compressor discharge pressure near compressor discharge flange. As the blow-off valve is slowly closed, the pressure will rise. Monitor both the pressure gage and the pressure transmitter. Watch for the pressure to bounce (see “d”). Also watch for any drop in pressure. At the first indication of a drop in pressure (with decreasing flow), consider this to be surge, and record data.
  f. Compressor flow rate. Watch for fluctuations in the flow meter differential pressure. Note that an electronic output on the flowmeter will not indicate surge unless the device is rated for dynamic conditions, with a rise time below 0.1 s. It is best to locally attach a manometer (for low pressure) or differential pressure gage and monitor this. Any dynamic differential pressure in excess of 20% of the nominal (steady state) differential at the given flow rate is to be considered surge, if no other indications (c, d, e, or f) are observed.
  g. Compressor vibration level. Pay particular attention to subsynchronous amplitudes. Very small increases or bouncing of amplitudes indicate possible onset of surge. An increase of 20% at the given speed of the overall vibration level, or 0.20 mils increase of the subsynchronous, while alone not a sign of surge, indicates the proximity of an instability. Use extra caution when exceeding these values.

(2) When any of these items (except the peak head condition, “e”) indicates surge, the position of the blow-off valve should be immediately noted, and then opened to the full open position.

(3) Close the valve back to within a few percent of the point where the instability occurred. Example: The suction pressure began to bounce at blow-off valve opening of 79%. The valve is immediately opened to 100% open. The blow-off valve is then closed back to 81% open.

(4) Wait an additional time period until data are stable and record data to assure the system is stable.

(5) Repeat steps 1 through 4 for the other speed lines, or inlet guide vane positions.

(6) Record all data for future reference.

Note that the ideal method of detecting the point of aerodynamic instability is to monitor dynamic pressure probes near the inlet to the impeller and in the diffuser. Flow instability can develop in either location. In some units, it appears in the inlet due to flow separation on the inlet of the impeller blades. The position of this point on the compressor head curve generally lines up with the point of peak head. On other units, stall will occur in the diffuser section. This is caused by the inability of the diffuser to overcome the compressor discharge pressure. This event may not fall in line with peak head.

Sophisticated instrumentation is not required to detect surge. The instability usually can clearly be heard and even felt when standing near the compressor. Sometimes, the instability is subtle and you must listen very closely.

If you are standing in the compressor discharge area, you may not hear an inlet stall condition. Likewise, if you are in the control room observing the flow and pressure on the slow-responding process monitor equipment, you may only see a deep hard surge condition, when it occurs.

Keep in mind that too much surging will eventually cause equipment failure. Surge the equipment hard enough and long enough and something will eventually break. When setting the surge line, the equipment should experience only one or two surge pulses. Allowing the unit to surge any more is only asking for trouble. In order to accomplish this, the recycle or blow-off valve must have a quick opening (1–2 s) response and a slow (30 s) closing time to keep the system stable.

Be safe and assume that the unit is very sensitive to surge and that the machine could easily wreck if surged very much.

Appendix G

Glossary of Terms

Absolute pressure The absolute pressure is the pressure measured above a perfect vacuum.

Absolute temperature The absolute temperature is the temperature measured above absolute zero. It is stated in degrees Rankin or Kelvin. The Rankin temperature is the Fahrenheit temperature plus 459.67 and the Kelvin temperature is the Celsius temperature plus 273.15.

Absolute viscosity Absolute viscosity is that property of any fluid that tends to resist a shearing force.

Adiabatic A process in which there is no heat transfer is called an adiabatic process.

Capacity The capacity of a compressor is the volume rate of flow, which is determined by delivered mass flow rate divided by inlet total density. For sidestream machines, this definition must be applied to individual sections.

Choke point The choke point is the point where the machine is run at a given speed and the flow is increased until maximum capacity is attained.

Control volume The control volume is a region of space selected for analysis where the flow streams entering and leaving can be quantitatively defined as well as the power input and heat exchange by conduction and radiation. Such a region can be considered to be in equilibrium for both a mass and energy balance.

Density Density is the mass of the gas per unit volume. It is a thermodynamic property and is determined at a point once the total pressure and temperature are known at the point.

Differential pressure The differential pressure is the difference between any two pressures measured with respect to a common reference (e.g., the difference between two absolute pressures).

Dimensional constant The dimensional constant, g_c, is required to account for the units of length, time, and force. It is equal to 32.174 ft-$lb_m/lb_f\ s^2$. The numerical value is unaffected by the local gravitational acceleration.

Equivalence The specified operating conditions and the test operating conditions are said to demonstrate equivalence when for the same flow coefficient the ratios of the three dimensionless parameters (specific volume ratio, Machine Mach number, and Machine Reynolds number) fall within prescribed limits.

Flow coefficient The flow coefficient is a dimensionless parameter defined as the mass flow rate divided by the inlet density, rotational speed, and the cube of the blade-tip diameter.

Fluctuation The fluctuation of a specific measurement is defined as the highest reading minus the lowest reading divided by the average of all readings expressed as a percent.

Fluid Reynolds number The fluid Reynolds number is the Reynolds number for the gas flow in a pipe. It is defined by the equation $\mathrm{Re} = VD/\nu'$, where velocity V is the average velocity at the pressure measuring station, D is the inside-pipe diameter at the pressure measuring station and the kinematic viscosity, ν' is that which exists for the static temperature and pressure at the measuring station.

Gage pressure The gage pressure is that pressure, which is measured directly with the existing barometric pressure as the zero base reference.

Gas power Gas power is the power transmitted to the gas. It does not include mechanical losses.

Isentropic compression Isentropic compression as used in PTC 10 refers to a reversible, adiabatic compression process.

Isentropic efficiency The isentropic efficiency is the ratio of the isentropic head to the gas work input.

Isentropic head Isentropic head is the work required to isentropically compress a gas from the inlet total pressure and total temperature to the discharge total pressure.

Isentropic work coefficient The isentropic work coefficient is the dimensionless ratio of the isentropic work to the sum of the squares of the blade-tip speeds of all stages in a given section.

Kinematic viscosity The kinematic viscosity of a fluid is the absolute viscosity divided by the fluid density.

Mach number The Fluid Mach number is the ratio of fluid velocity to acoustic velocity.

Machine Reynol ds number
The Machine Reynolds number is defined by the equation Rem $= Ub/\nu'$, where U is the velocity at the outer blade-tip diameter of the first impeller or of the first-stage rotor tip diameter of the leading edge, ν' is the total kinematic viscosity of the gas at the compressor inlet, and b is the characteristic length. For centrifugal compressors, b is the exit width at the outer blade diameter of the first-stage impeller. For axial compressors, b is the chord length at the tip of the first stage rotor blade.

Machine Mach number The machine Mach number is defined as the ratio of the blade velocity at the largest blade-tip diameter of the first impeller for centrifugal machines or at the tip diameter of the leading edge of the first stage rotor blade for axial flow machines to the acoustic velocity of the gas at the total inlet conditions.

Mechanical losses Mechanical losses are the total power consumed by frictional losses in integral gearing, bearings, and seals.

Polytropic compression Polytropic compression is a reversible compression process between the inlet total pressure and temperature and the discharge total pressure and temperature. The polytropic process follows a path such that the polytropic exponent is constant during the process.

Polytropic efficiency The polytropic efficiency is the ratio of the polytropic head to the gas work input.

Polytropic head Polytropic head is the reversible work required to compress a unit mass of gas by a polytropic process from the inlet total pressure and temperature to the discharge total pressure and temperature.

Polytropic work coefficient The polytropic work coefficient is the dimensionless ratio of the polytropic work to the sum of the squares of the blade-tip speeds of all stages in a given section.

Pressure ratio Pressure ratio is the ratio of the absolute discharge total pressure to the absolute inlet total pressure.

Pressure rise Pressure rise is the difference between the discharge total pressure and the inlet total pressure.

Ratio of specific heats The ratio of specific heats, k, is equal to c_p/c_v.

Raw data Raw data are the recorded observation of an instrument taken during the test run.

Reading A reading is the average of the corrected individual observations (raw data) at any given measurement station.

Reynolds number The Reynolds number is a dimensionless number that expresses the ratio of inertia forces to viscous force.

Section Section is defined as one or more stages having the same mass flow without external heat transfer other than natural casing heat transfer.

Shaft power (brake power) The shaft power (brake power) is the power delivered to the compressor shaft. It is the gas power plus the mechanical losses in the compressor.

Specified operating conditions The specified operating conditions are those conditions for which the compressor performance is to be determined.

Specific heat at constant pressure The specific heat at constant pressure is the change in enthalpy with respect to temperature at a constant pressure.

Specific heat at constant volume The specific heat at constant volume is the change in internal energy with respect to temperature at a constant specific volume.

Specific volume Specific volume is the volume occupied by a unit mass of gas. It is a thermodynamic property and is determined at a point once the total pressure and temperature are known at the point.

Specific volume ratio The specific volume ratio is the ratio of inlet specific volume to discharge specific volume.

Stage A stage for a centrifugal compressor comprises a single impeller and its associated stationary flow passages. A stage for an axial compressor comprises a single row of rotating blades and its associated stationary blades and flow passages.

Static pressure The static pressure is the pressure measured in such a manner that no effect is produced by the velocity of the flowing fluid.

Static temperature The static temperature is the temperature determined in such a way that no effect is produced by the velocity of the flowing fluid.

Surge point The compressor surge point is the capacity below which the compressor operation becomes unstable. This occurs when flow is reduced and the compressor back pressure exceeds the pressure developed by the compressor and a breakdown in flow results. This immediately causes a reversal in the flow direction and reduces the compressor back pressure. The moment this happens regular compression is resumed and the cycle is repeated.

Temperature rise Temperature rise is the difference between the discharge total temperature and the inlet total temperature.

Test operating conditions The test operating conditions are the operating conditions prevailing during the test.

Test point The test point consists of three or more readings that have been averaged and fall within the permissible specified fluctuation.

Total (stagnation) pressure The total (stagnation) pressure is an absolute or gage pressure that would exist when moving fluid is brought to rest and its kinetic energy is converted to enthalpy rise by an isentropic process from the flow condition to the stagnation condition. In a stationary body of fluid, the static and total pressures are equal.

Total (stagnation) temperature The total (stagnation) temperature is the temperature that would exist when a moving fluid is brought to rest and its kinetic energy is converted to an enthalpy rise by an isentropic process from the flow condition to the stagnation condition. In a stationary body of fluid, the static and the total temperatures are equal.

Total work input coefficient	The total work input coefficient is the dimensionless ratio of the total work input to the gas to the sum of the squares of the blade-tip speeds of all stages in a given section.
Velocity (kinetic) pressure	The velocity (kinetic) pressure is the difference between the total pressure and the static pressure at the same point in a fluid.
Velocity (kinetic) temperature	The velocity (kinetic) temperature is the difference between the total temperature and the static temperature at the measuring station.
Volume flow rate	The flow rate is the local mass flow rate divided by local density. It is used to determine volume flow ratio.
Volume flow ratio	The volume flow ratio is the ratio of volume flow rates at two points in the flow path.
Work	Gas work is the enthalpy rise of a unit mass of the gas compressed and delivered by the compressor from the inlet total pressure and temperature to the discharge total pressure and temperature.
Work input coefficient	The total work input coefficient is the dimensionless ratio of the enthalpy rise to the sum of the squares of the tip speeds of all stages in a given section.

This page intentionally left blank

References

[1] Bodger WK, Jensen RC. Fundamentals of fluid flow as applied to the design of axial flow compressors and fans. Syracuse, NY/Jeannette, PA: Carrier Corporation/Elliott Company; 1954.
[2] Wilson DG. The design of high-efficiency turbomachinery and gas turbines. Cambridge, MA: MIT Press; 1984.
[3] Sawyer RT. The modern gas turbine. New York: Prentice-Hall; 1945.
[4] Adams HT. Elements of internal combustion turbine theory. New Zealand: Canterbury University; 1949.
[5] Hill PG, Peterson CR. Mechanics and thermodynamics of propulsion. Reading, MA: Addison-Wesley; 1965.
[6] Vietmeyer N. They created the jet age. Pleasantville, NY: Reader's Digest; 1987.
[7] API Standard 617. Centrifugal compressors for general refinery service. 5th ed. Washington, DC: American Petroleum Institute; 1988.
[8] Baumeister T. Standard handbook for mechanical engineers. New York: McGraw-Hill; 1967.
[9] Sheperd DG. Principles of turbomachinery. New York: Macmillan; 1956.
[10] Paluselli DA, Basic aerodynamics of centrifugal compressors, Elliott Company, Jeannette, PA.
[11] Compressor refresher. Jeannette, PA: Elliott Company; 1975.
[12] Shames IH. Mechanics of fluids. McGraw-Hill; 2002.
[13] Van Wylen G, Sonntag R. Fundamentals of classical thermodynamics. New York: John Wiley and Sons; 1968.
[14] Derrickson GW. Thermodynamic review. Jeannette, PA: Elliott Company; 1968.
[15] The orifice meter. Pittsburgh, PA: Rockwell Manufacturing Company; 1938.
[16] Sassos M. Compressor components. Jeannette, PA: Elliott Company; 1986.
[17] Hallock DC. Centrifugal compressors … the cause of the curve. Jeannette, PA: Elliott Company; 1968.
[18] Salisbury R. Compressor performance. Jeannette, PA: Elliott Company; 1986.
[19] Whiteman P, Axial compressor design philosophy, Elliott Company, Jeannette, PA.
[20] Lapina R. Can you rerate your centrifugal compressor? Jeannette, PA: Elliott Company; 1975.
[21] ASME PTC 10. Power and test codes, compressors and exhausters. New York: American Society of Mechanical Engineers; 1997.
[22] Dunaway J. Guidelines for mechanical field testing of compressors. Jeannette, PA: Elliott Company; 1979.
[23] Hackel R, King R. Centrifugal compressor inlet piping—a practical guide. Jeannette, PA: Elliott Company; 1977.
[24] Bensema D. Field performance testing. Jeannette, PA: Elliott Company; 1986.
[25] Leipman HW, Roshko A. Elements of gas dynamics. New York: John Wiley and Sons; 1957.
[26] Lock JA. Techniques for more accurate centrifugal compressor performance evaluation. Southwest research, turbomachinery symposium, Houston, TX; 1981.
[27] Ishida M, Senoo Y. The pressure losses due to the tip clearance of centrifugal blowers. New York: American Society of Mechanical Engineers; 1980.
[28] Lakshminarayana B. Methods of predicting the tip clearance effects in axial flow turbomachinery. New York: Journal of Basic Engineering, American Society of Mechanical Engineers; 1970.
[29] ASME PTC 19.5. Fluid meters. New York: American Society of Mechanical Engineers; 1971.
[30] Bulletin P-25C. Elliott multistage centrifugal compressors. Jeannette, PA: Elliott Company; 1985.
[31] Engineering data book. Tulsa, OK: Natural Gas Processors Suppliers Association; 1967, 1972.
[32] Properties of commonly used refrigerants. Washington, DC: Air Conditioning and Refrigeration Institute; 1957.
[33] Wilcox C, editor. Practical methods for field performance testing centrifugal compressors. Proceedings of the 28th turbomachinery symposium 1999.
[34] Annubar® flow handbook. Boulder, CO: Dieterich Standard; 1997.
[35] Aungier RH. Centrifugal compressors: a strategy for aerodynamic design and analysis. New York: ASME Press; 2000.
[36] PTC-22. Gas turbine power plants, performance test code. ASME; 1985.
[37] North american combustion handbook. Cleveland, OH: North American Manufacturing Company; 1965.

FURTHER READING

[1] Church AH. Centrifugal pumps and blowers. New York: John Wiley and Sons; 1967.
[2] Cotton KC. Evaluating and improving steam turbine performance. Rexford, NY: Cotton Fact Inc.; 1998.
[3] Fergerson TB. The centrifugal compressor stage. London: Butterworths; 1963.
[4] Flow of fluids through valves, fittings, and pipe. Crane technical paper no. 410, Crane Co., New York; 1978.
[5] Fluid Flow Data Book. Schenectady, NY: General Electric Corporate Research and Development; 1975.
[6] Kunkle J, Wilson S, Cota R, editors. Compressed gas handbook. Washington, DC: NASA; 1969.
[7] AMCA 210-74 and ASHRAE 51-75. Laboratory methods of testing fans for rating. New York: American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc.; 1974.

[8] Lapina RP. Estimating centrifugal compressor performance. Houston, TX: Gulf Publishing; 1982.
[9] Lee BI, Kesler MG. A generalized thermodynamic correlation based on three-parameter corresponding states. Mobil Research and Development Corp. AICHE J; 1975.
[10] Nisenfeld AE. Centrifugal compressors—principles of operation and control. Research Triangle Park, NC: Instrument Society of America; 1982.
[11] Pichot P. Compressor application engineering. Houston, TX: Gulf Publishing; 1986.
[12] Spink LK. Principles and practice of flow meter engineering. Foxboro, MA: The Foxboro Company; 1967.
[13] Starling KE. Fluid thermodynamic properties for light petroleum systems. Houston, TX: Gulf Publishing; 1973.
[14] Walas SM. Phase equilibria in chemical engineering. Stoneham, MA: Butterworths; 1985.
[15] Walsh PP, Fletcher P. Gas turbine performance. Fairfield NJ: ASME Press; 2004.

Compressor and Turbine Design Software Performance-Monitoring Software

Remote-Monitoring Services

Software is available for your IBM-PC compatible computer for performance calculations. These programs handle compressor and steam turbine design and selection as well as shop and field performance monitoring and testing. Some functions include compressor and steam turbine roughout, gas properties, head, flow, and efficiency calculations plus plotting of results. The design software is a good tool to design new equipment from scratch or model existing equipment to predict new conditions or complete a rerate. Performance-monitoring software can connect to plant database and automatically and continuously process data for continuous trending.

Asset web monitor (AWM) service provides continuous remote monitoring of performance and vibration and technical assistance for problem resolution.

For order details or for further information on program & service capabilities, please contact:

This page intentionally left blank

Index

Note: Page numbers followed by *f* indicate figures, *t* indicate tables, and *b* indicate boxes.

Compressor Performance

Aerodynamics For The User

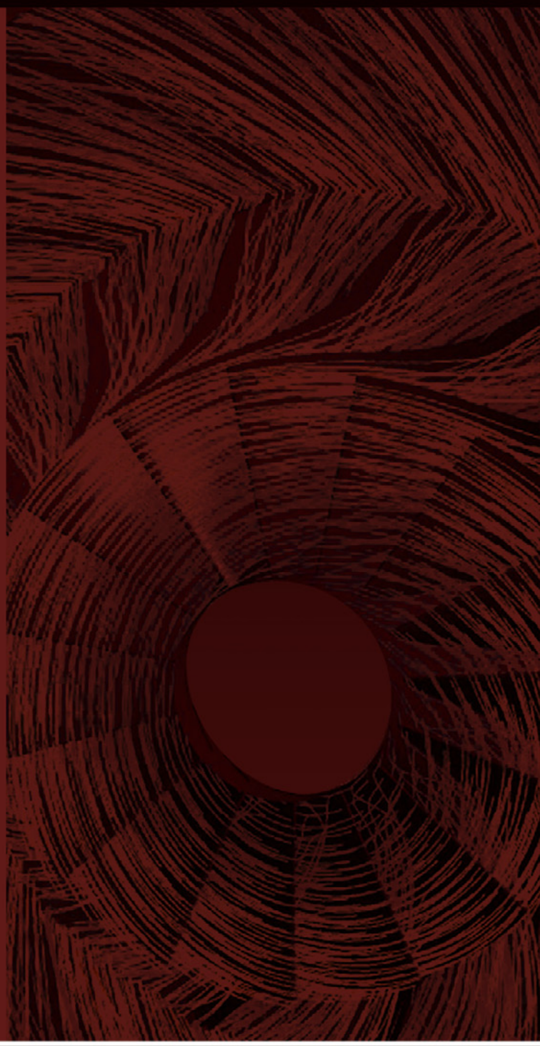
Third Edition **M. Theodore Gresh**

Compressor Performance has been a trusted reference book for compressor design and maintenance for over 25 years, and is relied upon by engineers and students worldwide.

This new edition is updated throughout to cover new regulations and technology relevant to compressors, and new content adds coverage of strings of equipment, including gas turbines.

Part I will present the theory behind compressor performance, while Part II will discuss the various applications demonstrated by case studies.

The book covers the full spectrum of information needed for an individual to select, operate, test, and maintain axial or centrifugal compressors. Basic aerodynamic theory provides users with the "how's" and "why's" of compressor design, troubleshooting guidelines help maintenance engineers save time in the field, and many example problems and reference data such as gas properties and flow meter calculations enable easy analysis of compressor performance in practice.



M. Theodore Gresh is president of Flexware, Inc., Grapeville, PA, United States. He has been involved in the design of high-efficiency centrifugal compressor staging, field testing of compressors and steam turbines, troubleshooting various field problems, including performance problems, rotor dynamics issues, impeller failures, and seal problems for over 40 years.

ELSEVIER

Butterworth-Heinemann
An imprint of Elsevier
elsevier.com/books-and-journals

ISBN 978-0-12-814219-6